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AIRPLANE ENGINE DESIGN

by

Charles F. Park

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AIRPLANE ENGINE DESIGN

Syllabus of Lectures

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AIRPLANE ENGINE DESIGN

Description of Course. Subject matter, available material and precedent.

The Internal Combustion Engine. Its development and influence.

Airplanes became possible when this engine was sufficiently developed.

Essential difference of this engine and the steam engine; relative efficiency.

Difficulties to be encountered, high temperatures and residue of combustion; limited range of fuels that can be used.

What we must know to design a motor.

Action of the Internal Combustion Engine. Relies on the heating of air by means of fuel burned within the cylinder. The combustion causes a rapid rise of temperature with consequent rise of pressure which is utilized on the moving piston.

Two Distinct Methods,-

1. "Explosion" Engines or "Constant Volume" Engines.
2. "Diesel" Engines or "Constant Pressure" Engines.

Method of working of each.

"Semi-Diesel" engines, a compromise between the two; a promising future.

Essential difference between the two classes.

The Explosion Engine. Various cycles.

Two-stroke cycle, Four-stroke cycle, Six-stroke cycle.

Advantages and disadvantages of each.

Four-Stroke cycle best adapted for airplane service.

Thermal Efficiency. Dependent mainly on degree of compression.

Theoretical efficiency, a standard set-up with which actual efficiency of the engine can be compared.

Chemical combustion of fuel; and chemical action.

"Lower heating value" of a fuel is the heating value after the heat required for the conversion of the water into steam has been deducted.

The ratio of the heat converted into work to the total heat available (lower heating value) is the thermal efficiency.

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Review of Fundamentals

Specific Heat of Air

Specific heat of air at constant pressure is 0.2375 B.T.U.
Specific heat of air at constant volume is 0.1691 B.T.U.
The difference of these two specific heats, 0.0684 B.T.U., is the amount of heat which is used to effect the expansion under constant atmospheric pressure, same amount of heat would be liberated if the air were compressed to its original volume.

The ratio between the two specific heats of air is

$$n = \frac{0.2375}{0.1691} = 1.4 \text{ nearly.}$$

Boyle's Law. At constant temperature, PV = constant.

Charles' Law. $\frac{P}{P_1} = \frac{T}{T_1}$ and $\frac{V}{V_1} = \frac{T}{T_1}$

The pressure or volume of a gas is proportional to its absolute temperature.

Absolute Zero of Temperature.

-273°C or - 459°F.

Thermal Efficiency

$$E = \frac{T - T_1}{T} = 1 - \frac{T_1}{T}$$

The ratio of the heat supplied minus the heat rejected to the available supply of heat.
Efficiency could only reach 100% by reducing the temperature at the end of the cycle to absolute zero.
The lowest temperature cannot be much reduced, and it is subject to not much variation. So, the efficiency can only be materially increased by increasing the maximum temperature.
Illustrative example.

Isothermal and Adiabatic Expansion.

Expansion and compression at constant temperature is "isothermal", PV = constant.

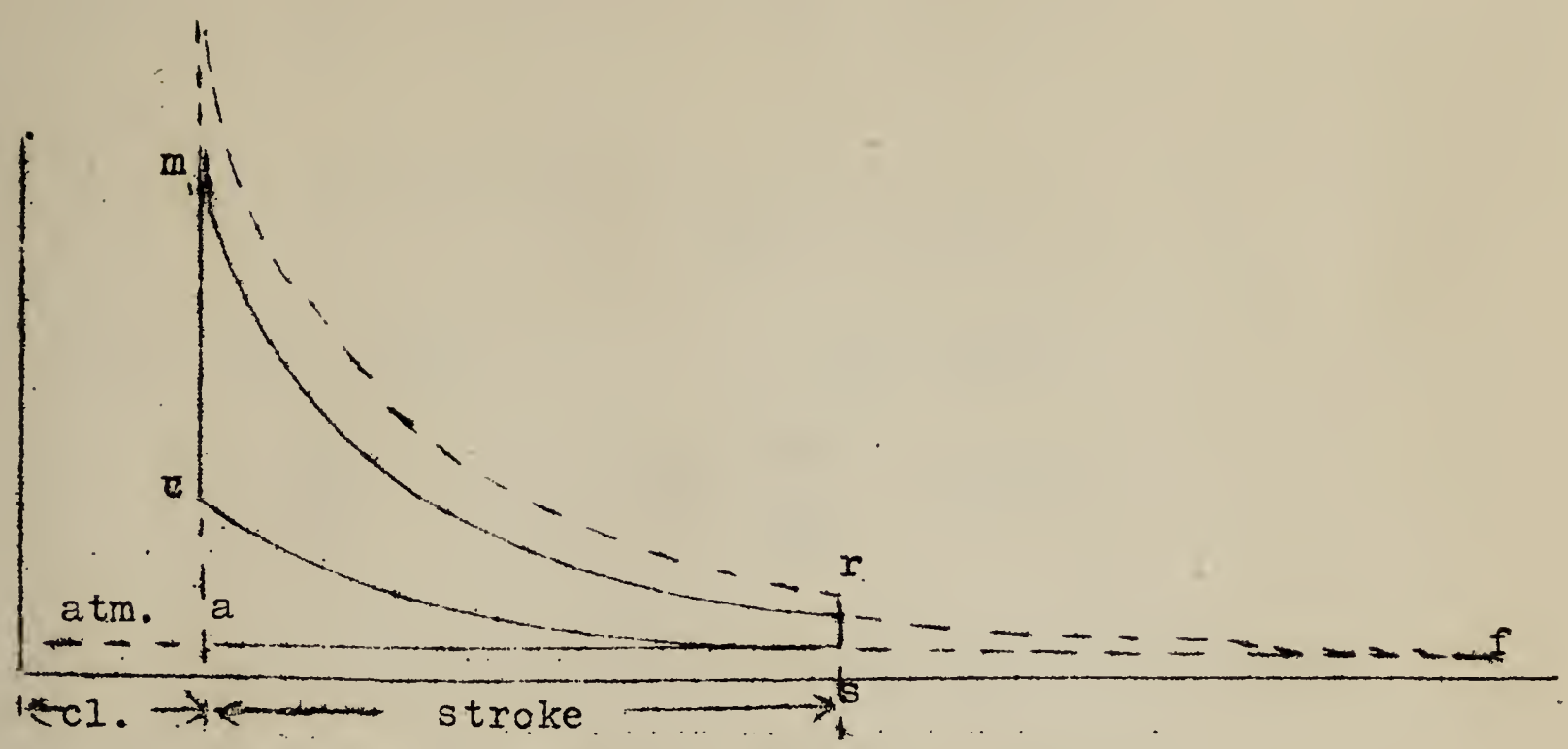
If no heat is allowed to escape, the temperature not remaining constant, the expansion and compression is "adiabatic". PV^n = constant. when "n" is ratio of specific heats.

$$\frac{T_1}{T} = \left(\frac{V}{V_1}\right)^{n-1}$$

Illustrative example.

In an internal combustion engine the expansion and compression are more nearly adiabatic than isothermal, and dependent on the value of "n" for the explosive mixture under the working conditions.

IDEAL INDICATOR DIAGRAM



$$\text{Clearance} = \frac{\text{vol. of clearance space}}{\text{vol. of piston displacement}}$$

$$\text{Compression ratio} = \frac{\text{vol. of clearance + vol. of piston displacement}}{\text{vol. of clearance}}$$

<u>Method of Finding Theoretical Pressures and Temperatures.</u>

Example: Take clearance = 25%

Compression Ratio = 5.

Initial pressure, p_a = atmospheric, 14.7 abs.

Initial temperature t_a = 60°F = 519° abs.

Then $P_s = P_a = 14.7$ $\qquad T_s = T_a = \underline{519°}$

$$\frac{P_c}{P_s} = \left(\frac{V_s}{V_c}\right)^{1.4}$$

$$P_c = 14.7 \times 5^{1.4} = \underline{140} \text{ pounds abs.}$$

$$\frac{T_c}{T_s} = \left(\frac{V_s}{V_c}\right)^{0.4} \qquad T_c = 519 \times 5^{0.4} = \underline{988}.$$

Heat of explosion = weight of charge X its specific heat X rise of temperature.

$$80\,(V_s - V_a) = 0.0784\ V_s \times 0.169\ (T_m - T_c)$$

$$T_m - T_c = 80 \times \frac{4}{5} \times \frac{1}{.0784 \times .169}$$

$$T_m - 988 = 4720 \qquad T_m = 5708°$$

$$\frac{P_m}{P_c} = \frac{T_m}{T_c} \qquad P_m = 140\ \frac{5708}{988} = 808.8$$

No such values for T_m and P_m are ever actually reached.

Let us assume

$$P_m = \underline{450}\ \text{abs.} \qquad \text{Then,}$$

$$T_m = 988\ \frac{450}{190}, \qquad T_m = \underline{3176°}\ \text{abs.}$$

$$P_r = 450\ \left(\frac{1}{5}\right)^{1.4} = \underline{47.3}$$

$$T_r = 3176\left(\frac{1}{5}\right)^{0.4} = \underline{1668°}$$

<u>Thermal Efficiency</u>

Ratio of heat accounted-for as doing work on the piston to the total heat supplied to the engine.

Heat supplied = specific heat X $(T_m - T_c)$

Heat discharged = specific heat X $(T_r - T_s)$

$$E = \frac{\text{Spec. heat } (T_m - T_c) - \text{Spec. heat } (T_r - T_s)}{\text{Spec. heat } (T_m - T_c)}$$

$$= 1 - \frac{T_r - T_s}{T_m - T_c} = 1 - \frac{1688 - 519}{3176 - 988} = .475$$

Since the heat is both supplied and discharged at constant volume, and since both the compression and expansion are adiabatic, the efficiency is dependent solely on the compression ratio, and is independent of the maximum or initial temperature. Therefore,

$$E = 1 - \left(\frac{V_c}{V}\right)^{n-1} = 1 - \left(\frac{1}{5}\right)^{0.4} = 47.5\%$$

This is known as the "Air Standard" Efficiency.

<u>Prolongation of Expansion Stroke</u>. Increase of expansion ratio in relation to compression ratio.

If expansion were continued to atmospheric pressure,

$$\left(\frac{P_m}{P_f}\right)^{\frac{1}{n}} = \left(\frac{450}{14.7}\right)^{\frac{1}{1.4}} = 11.5 \text{ expansions}$$

Then, length of expansion stroke must be $\frac{11.5}{5}$ or 2.3 times the length of the compression stroke.

$$T_f = T_m\left(\frac{1}{11.5}\right)^{n-1} = 1197.$$

$$E = 1 - n\left(\frac{T_f - T_s}{T_m - T_c}\right) = 1 - 1.4\left(\frac{1197 - 519}{3176 - 988}\right) = 56.7\%$$

In such case, the air standard efficiency would be increased from 47.5% to 56.7%; but the efficiency would no longer be dependent on the compression ratio alone, but would vary according to the maximum temperature. The length of the expansion stroke would also be dependent on the maximum pressure and temperature; and it would be necessary, for best results, to vary the length of the expansion stroke in relation to the compression stroke for every variation of the temperature and pressure, which is impracticable in ordinary type of engine.

Conclusions

The air standard efficiency, when the gases are expanded to the same volume as before compression, is dependent solely on the ratio of compression.

The efficiency; when the expansion is carried to atmospheric pressure, is no longer dependent on the ratio of compression, but is also dependent on the maximum temperature, and increases slightly with increase of temperature.

These conclusions are based on the assumptions,-

That the working fluid is pure dry air.
That its specific heat remains constant at constant volume for the whole range of temperature.
That both the compression and expansion are adiabatic.
That combustion is instantaneous and takes place at constant volume; and
That the gases are expanded to precisely the same volume as they originally occupied.

In actual practice none of these assumptions hold good, for:

The working fluid is not pure dry air, but contains a proportion of other gases whose specific heat at constant volume and constant pressure do not bear the same relation as in the case of air.

It has been proven recently that the specific heat of the working fluid is not constant over a wide range of temperature, but increases considerably at very high temperature. This is of great importance.

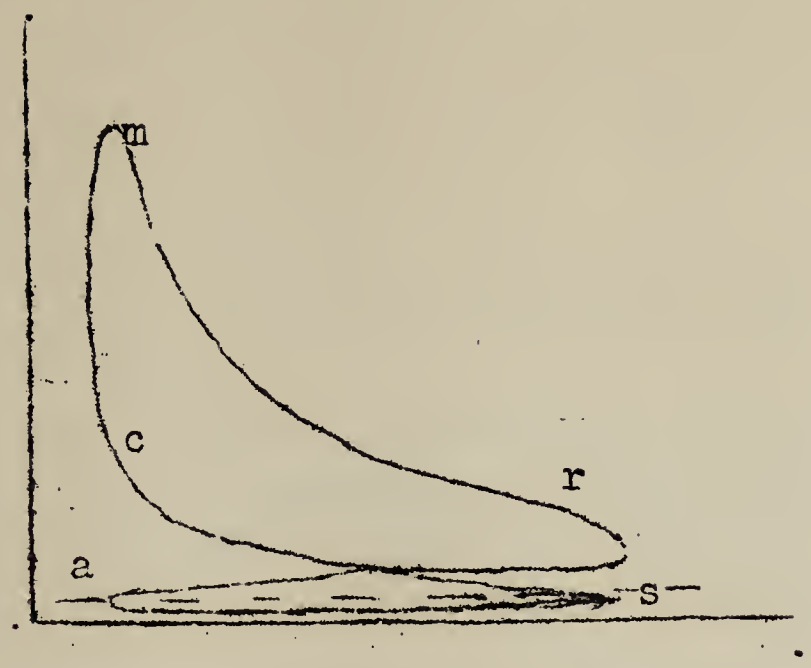
Neither the compression not expansion is truly adiabatic, owing to the change that takes place in the specific heat. Also heat is lost to the cylinder walls.

The combustion is not instantaneous; the propagation of flame throughout the whole of the working fluid takes an appreciable amount of time, and combustion continues during part of the expansion stroke.

The gases are not expanded to the same volume as they occupied before compression, for it is necessary to open the exhaust valve slightly before the end of the stroke.

To make due allowances for all these discrepancies renders the calculation of the thermal efficiency of an engine exceedingly complicated and difficult. Although the air standard efficiency of explosive engines is independent of the maximum temperature, it will be found that the lower the maximum temperature the higher the efficiency. This is a matter of great importance.

REAL INDICATOR DIAGRAM



Differences between actual and ideal engine occur in each part of the cycle.

Admission Stroke,- Pressure below atmospheric, necessary to cause charge to flow into cylinder with sufficient velocity,- may be 12 to 14 pounds. Charge heated by residual burned gas and by contact with hot piston,- temperature may be 160° - 260° F.

Compression Stroke,- Not exactly adiabatic,- "n" may be 1.23 to 1.33; pressure may be 100 to 120 pounds.

Explosion,- Neither instantaneous nor complete; slow with weak mixtures; vitiation of charge has delaying effect.

Weak Mixtures

Rise of temperature and pressure may be only 3/5 that of ideal engine; temperature may be 2000° to 3000° F., pressure may be 250 to 400 pounds; raising compression increases each. Power considerably less than ideal. Water jacket may absorb 40% of total heat of combustion.

Working Stroke,- Expansion curve not exactly adiabatic,- "n" may be 1.25 to 1.5; cylinder walls give-back heat, especially when mixture is weak.

Exhaust Stroke,- Exhaust begins before end of working stroke, diagram rounded off. Pressure at release may be 35 to 75 pounds, temperature may be 1200° to 1800° F.; pressure during stroke may be 15.5 to 17 pounds. Efficiency less than ideal.

Mean Effective Pressure,- Diagram consists of two loops, upper represents positive work, lower negative work, difference of two areas is total work on piston; smaller is pumping work, a definite loss; planimeter gives difference.

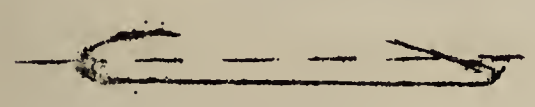
Causes of the Different Losses

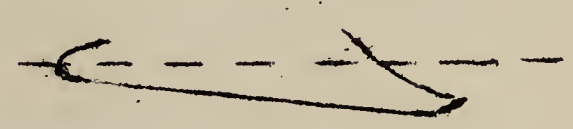
Suction Stroke,- Power used-up by piston suction corresponding to amount required to overcome,-

1. Resistance in carburetor. 2. resistance in inlet manifold and passages, 3. resistance of the inlet valve.

This loss should be small.

Throttling the engine gives a loss which would be shown by a lowering of the suction line. The area below the atmospheric line represents the power los t during suction stroke. If the velocity of gas through the valve passage is constant, the exhaust line will be horizontal. If the valve passage is too small, the suction line will decline.





To obtain high compression, the charge should be kept cool before entering the cylinder. Preheating for more through vaporization is bad for compression and the amount of charge, but necessary for the purpose named. A cooler

charge will often result in more power, even though the fuel efficiency is lower. The higher the compression pressure, the higher the efficiency; and the greater the weight of charge, the greater the power. The weight of charge decreases with temperature rises.

Important to have perfectly combustible mixture, - just the right proportion of air and gasoline vapor. One proportion for which flame propagation is highest. Slight deviation therefrom will make marked difference in velocity of propagation, power and thermal efficiency.

Compression Stroke,- Cylinder not completely filled.

Proportion of residual gas.

Character of compression curve depends on,-

Temperature of walls; the cooler they are the nearer the curve approaches isothermal.

Time taken for compression; the higher the speed, the higher the compression pressure.

If compression pressure is lower at higher speeds it is due to increased loss of charge or suction, which means increased fluid friction through valve passage,- cylinder does not fill. Importance of large valves and passages.

The higher the r.p.m., the less heat lost to the walls; but the larger the ratio of surface to volume of compression space the more heat will be lost.

Volume increases as cube of linear dimensions; the surface increases as the square.

Effect with multi-cylinder engines.

Throttling the engine will reduce the compression, retard combustion, lower the height of the peak and the efficiency.

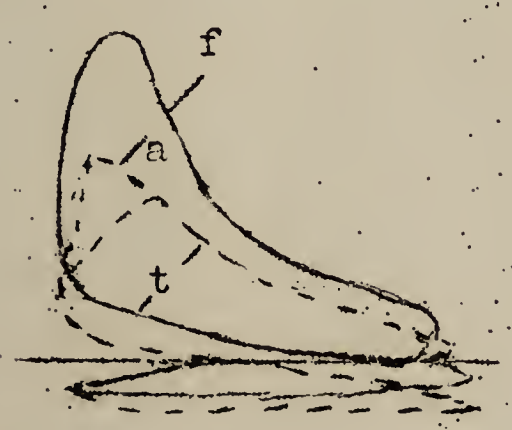
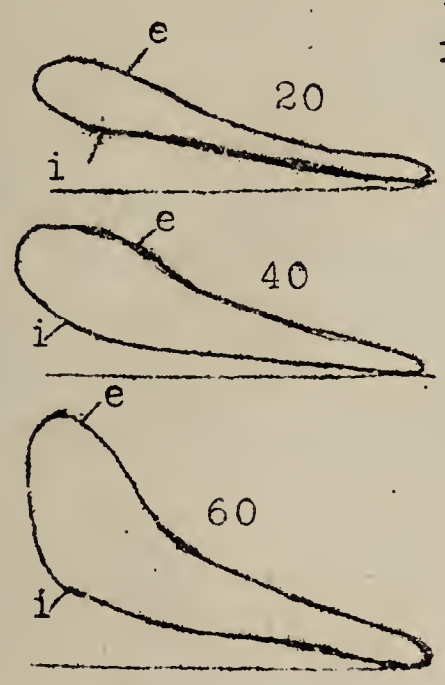


Diagram for throttling is "t". By advancing ignition area of diagram could have been increased - line "a" Peak would be raised and power increased by advancing spark lever with corresponding changes in throttle lever.

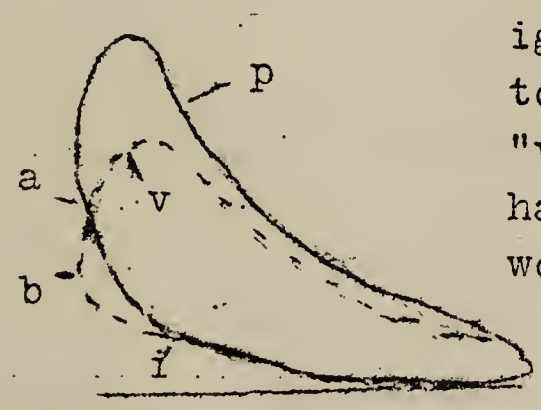
Point above which an increase of compression will not give more power, owing to increased losses on compression and heat lost through cylinder walls. Greatest drawback to high compression is increased temperature, - pre-ignition.

High compression and pure charges have important bearing on rapidity of combustion.



The diagrams 20, 40, and 60 show different flame propagations due to the different compression pressures of 20, 40 and 60 pounds. Point of ignition "i" was shifted; "e" is the point where the total charge is practically ignited and expansion begins.

Diagrams "p" and "v" show the effect of purity charge. "p", pure charge; "v" charge vitiated with large amount of burned gases. Compression and point of ignition the same in each. After ignition at "i", the curve "p" rises to "a" in the same time that curve "v" rises to "b". If the ignition had been advanced for "v" its peak would have been higher, as pressure "b" at end of stroke would have been higher, giving a higher temperature and more rapid flame propagation.



When the amount of burned gas in the cylinder is changed the ignition point also should be changed.

Time of Ignition and the Working Stroke,- The higher the compression, the smaller the quantity of gasoline required to obtain the best results, and the later the point of ignition. If the point of ignition is not advanced when the charge consists of an excess of air or an excess of gas, there will be additional delay in the flame propagation besides reduction in height of the peak.

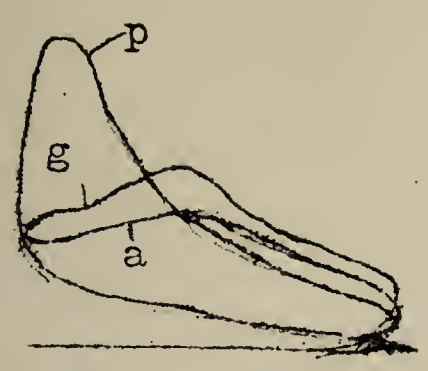
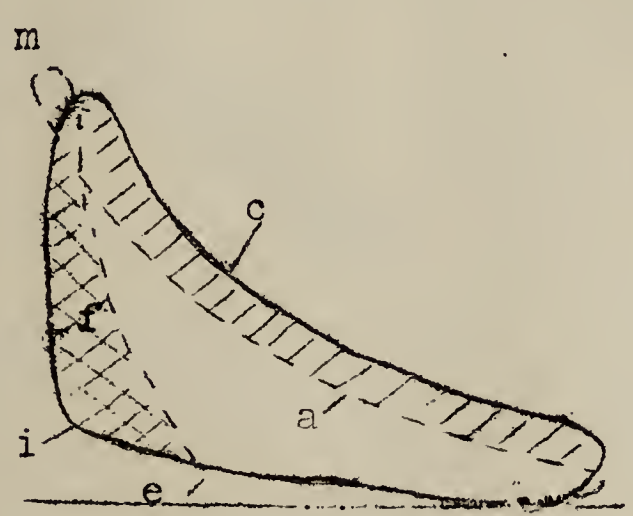


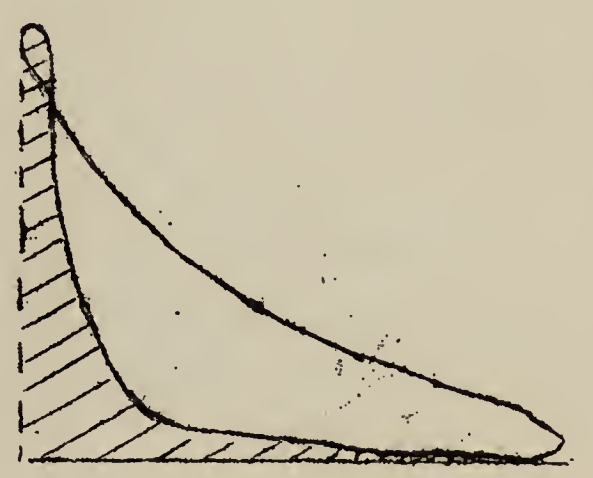
Diagram "p" is for perfect mixture; diagram "a" for an excess of air, "g" for an excess of gas.
When the mixture is imperfect, its temperature will have to be increased before ignition of the total charge can take place; increased time required for combustion of the total charge. Diagram "g" shows that some layers of the rich gas are ignited late; curve does not follow adiabatic expansion, pressure and temperature at release are high, engine heats-up.

Important to have point of ignition correct when mixture is perfect. If ignition is too early combustion will begin before dead center, increasing negative work and lowering expansion line and power of engine.

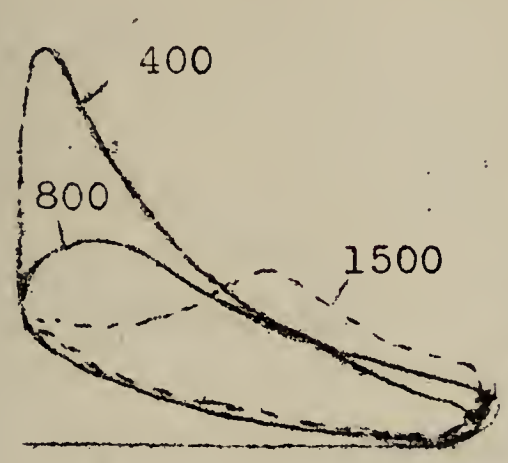


Ignition point, "i" is correct in diagram "c". In diagram "a" ignition, "e", is advanced too much. Loss in power is represented by the shaded area.

Area "f" indicates the power opposing the motion of the piston. If this area added of that under the compression line is much more than 85% of remaining area it may stop the engine at no load, remaining 15% being power loss in suction stroke and friction.

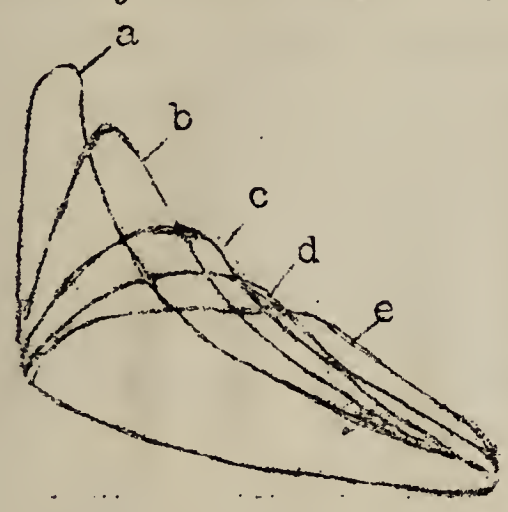


Flame propagation is not instantaneous; varying piston speeds require different points of ignition to give maximum pressure at end of stroke.



Diagrams are for same point of ignition when speeds are 400, 800 and 1500 r.p.m. About same kind of diagrams if speed were constant and point of ignition retarded. Pressure and temperature at release higher with the 800 and 1500; late ignition heats-up engine.

The five diagrams show the effect on combustion of excessive cylinder wall cooling. Five different cylinder wall



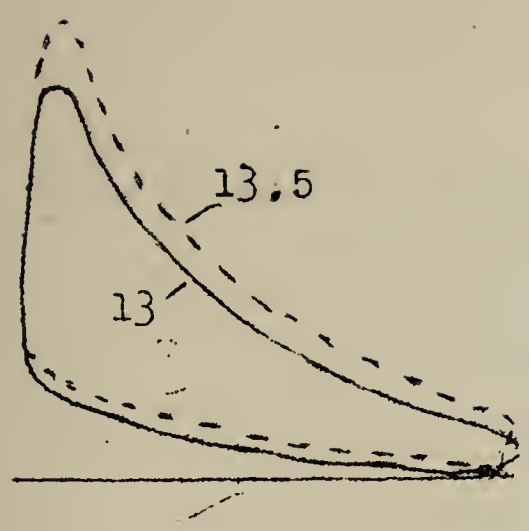
temperatures; "a" hottest, "e" coldest. Also, the greater the wall surfade the lower the peaks. By advancing the ignition more power would have been obtained, peaks would have been higher. Proper point of ignition depends on cylinder wall temperature. Reverse effect when cold engine is started.

Compression is the same in the five cases; amount of charge is larger when walls are cold, gain in charge neutralizes heat loss through jacket. Higher peaks and more power by keeping walls at low temperature; but thermal efficiency is lower, losses through walls is greater.

Expansion Stroke. Height of expansion curve depends on:

Volume of the charge
Quality of the working mixture
Compression pressure
Point of ignition
Speed of engine
Losses through cylinder walls.

A small change in pressure at end of suction stroke or beginning of compression stroke makes astonishing change in expansion line,- An increase of only a half pound produces large increase in power.



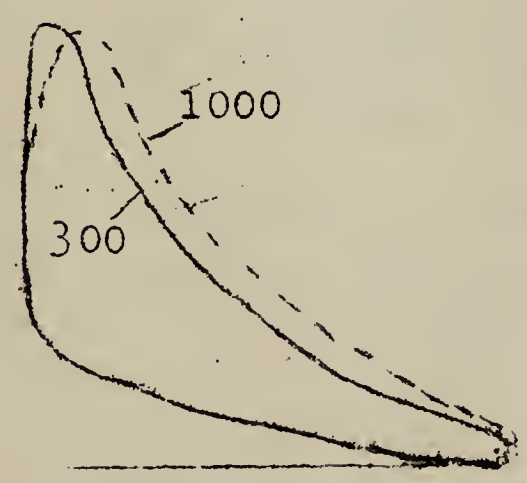
The two diagrams are for the best mixture and best point of ignition at same piston speed. One with compression beginning at 13 pounds, and the other at 13.5 pounds. Chief reason for better expansion curve and increased power is increased volume of charge.

Increase in size of valves and passages gives larger volume of charge, but the increase in size means larger valve pockets which carries with it larger heat losses.

The larger the area of cylinder walls, the larger the heat losses; these losses may be reduced by increasing the wall temperature. If temperature is too high, difficulties from lubrication and friction arise.

For highest efficiency, difference in temperature of entering and leaving cooling water should be a minimum.

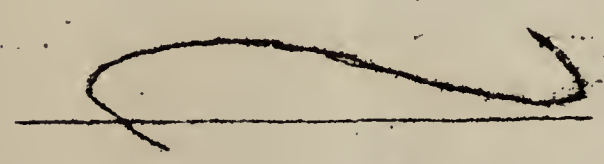
Length of time the hot gases are in contact with cylinder walls affects thermal efficiency.



The two diagrams taken under most favorable conditions of ignition and charge. One at 300 r.p.m., the other at 1000 r.p.m. Compression the same in each case, but ignition advance for 1000 r.p.m.

At higher engine speeds the gas expands at higher pressure, because of smaller thermal losses.

Exhaust Stroke. Always a certain loss due to back pressure. If exhaust valve is too small or its lift insufficient, or the passages or exit restricted there will result an ascending exhaust line giving a back-pressure loss which should be corrected.



The pressure at the end of the exhaust stroke is an index of the amount of burned gases remaining in clearance space; any means of reducing this amount will result in increase of engine power.

SOURCES OF LOSS OF EFFICIENCY

Cooling Water Losses. Besides increase of specific heat at high temperatures, loss of heat to walls and piston account for a substantial proportion of the discrepancy. These parts should be kept comparatively cool. Considerable proportion of heat liberated during combustion and expansion pass into cylinder walls and is lost. In order to reduce these losses it is essential that,-

1. Surface exposed to gases should be as small as possible.
2. Temperature of working fluid should be as low as possible.

Heat lost to cylinder walls not easily determined. True loss generally ranges between 40% and 60% of that accounted by the cooling water, depending on design, size and speed.

Slow Combustion. Loss due to incomplete or delayed combustion. Incomplete combustion is generally due to,-

1. Stagnation of the working fluid.
2. Presence of pockets or recesses in combustion space.
3. Presence of large proportion of inert gases.

Stagnation of working fluid one of the most important. For complete combustion it is essential that the gases be in a state of rapid motion; normal rate of propagation of flame is too slow. If velocity of gases at time of combustion is high, propagation of flames may be rapid.

Presence of large proportion if inert gases delays rate of flame propagation only when intimately mixed with mixture. If mixing can be prevented their presence may be desirable, owing to lower temperature.

Practical Limits of Compression. Extremely high compression ratios not possible, because of pre-ignition. Compression ratio must suit the fuel used. The richer the gas the lower the ignition temperature, and therefore the lower the safe compression ratio. Exceptions.

Compression ratios of from 4 to 5 : 1 are common. Other causes of premature ignition are:

1. Presence of uncooled parts of the engine, exhaust valves, etc.
2. Presence of coating of carbon.
3. Presence of detached particles of carbon, igniter electrodes, etc. insulated from water jacket.
4. Detonation, occurring when working fluid is compressed beyond certain point.

If detonation can be prevented higher compression ratios can be used. Important that contents of cylinder at beginning of compression stroke be at lowest possible temperature.

Lower compression ratios are employed than would appear necessary.

Power and Cylinder Capacity. The power obtained from a given size of cylinder depends on the weight of air that can be passed through the engine, and on the efficiency with which it can be burned; sufficient fuel can always be admitted to combine with it. To be capable of dealing with the maximum weight of air,-

1. The piston speed must be as high as possible.
2. The valves and passages must offer minimum obstruction to flow.
3. The incoming charge must be at as low a temperature as possible.

Volumetric efficiency must be as high as possible. Conditions also for high efficiency. To obtain highest possible powers high temperatures are employed which are not compatible with high thermal efficiency. Volumetric efficiency depends largely on the valves,- their size, position, and time of opening and closing, also design of passages leading to and from them. Advantage should be taken of inertia of the gases; high velocity of entry maintains them in state of turbulence.

Low temperature of charge can be met only by reducing the proportion of residual exhaust gases to a minimum. Accomplished by,-

1. Perfectly free outlet, proper timing of exhaust valve, taking advantage of inertia of gases in exhaust pipes.
2. Employing some method of scavenging.

Mechanical Efficiency. Ratio between indicated power and brake or available power at the crank shaft is the mechanical efficiency; ranges between 80% and 90%. Indicated thermal efficiency is generally within 10% to 20% of theoretical.

To obtain highest possible mechanical efficiency:

1. Ratio of maximum to mean pressure must be as low as possible.
2. Piston speed and rotative speed must not be high.
3. Weight of reciprocating parts must be a minimum.
4. Mean effective pressure must be as high as possible.
5. Moving parts must be in perfect static and dynamic balance.

1. Influence of Maximum and Mean Pressure Ratios. Should be low because the parts of the engine have to be strong enough to withstand the maximum pressures. If ratio is high, the weight, inertia and friction of moving parts is excessive. Compatible with thermodynamic condition, but not with high compression ratio.
2. Revolutions and Stroke. Piston speed and rotative speed should be low; the greater part of friction is due to inertia of reciprocating parts, which increases as piston speed and as the square of rotative speed. Necessary to employ high piston speed to obtain reasonable power and to reduce heat losses to a minimum. The two conditions are conflicting; compromises. Desirable to increase piston speed without increasing rotative speed, - long stroke.
3. Weight of Reciprocating Parts. Friction due to inertia,- may be as much as 70% of total mechanical friction.
4. Mean Effective Pressure. Must be as high as possible. Illustrations.
5. Balance. A difficulty with most engines. Power dissipated in vibration.

Manufacturing Cost. To be a practical success, cost per horsepower must not be too high.

Conditions for Low Cost of Production.

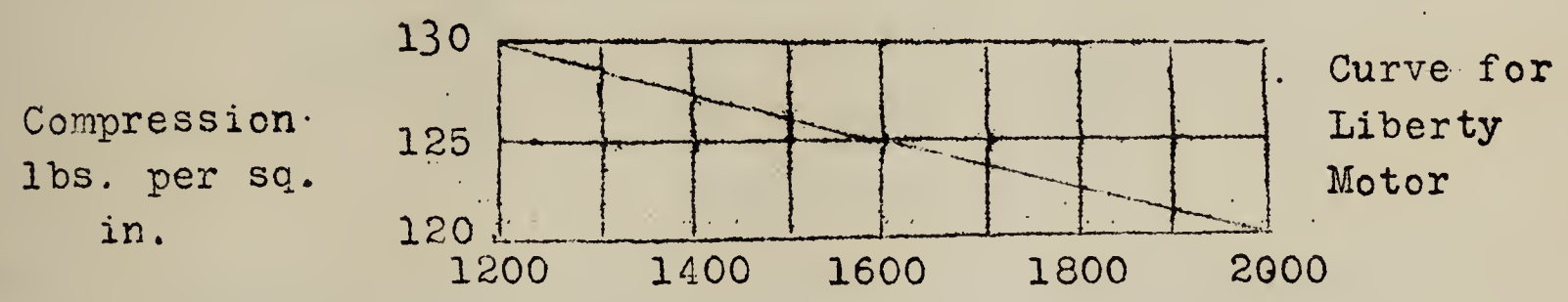
1. Mean effective pressure as high as possible.
2. Maximum pressures as low as possible.
3. Stroke should be short.
4. Piston speed as high as possible.
5. Valve gear simple, simple form of operating.
6. No intricate parts. Degree of accuracy that is ordinarily attainable.

1. The higher the mean pressure the greater the power for a given size of engine. Condition for good mechanical efficiency, but not for best thermal efficiency.
2. The lower the maximum pressure, the lighter and cheaper the engine. Condition for good thermal and mechanical efficiency.
3. Cost depends largely on length of stroke. Increase in stroke necessitates increase in many parts of the engine. Long stroke engine more bulky and heavy. Short stroke not compatible with high thermal or mechanical efficiency; compromise.
4. Power varies as the piston speed. Increase in piston speed increases power without increasing cost. High piston speed combined with short stroke the worst condition for mechanical efficiency. Conflicting conditions; compromise.
5. From thermodynamic point combustion chamber should be hemispherical, no recesses. Inclined valves expensive; other arrangements simple and inexpensive.
6. Free from intricate parts. Standard of workmanship and accuracy has improved. Standardization.

Resumé

Large compression ratios have been attempted to obtain highest power for given cylinder size. Increase of compression pressure beyond certain amount gives loss of power.

Higher compressions should be usable in high-speed engine than in low-speed one. Not always so in practice. Compression pressure tends to fall with an increase of engine speed in consequence of increase of suction at the moment the inlet valve closes.



Several causes for loss of power at high compressions. Loss of heat through cylinder walls is larger; shape of combustion chamber less suitable. Explosion prevented from developing its full pressure.

Practical objections to very high compressions. Effect of leakage more important; friction and wear and tear intensified. Inequality of turning moment.

To make an engine as small as possible for its horsepower, a rather high compression would be used; if more important to make engine run smoothly and quietly a lower compression would be employed. Loss of power by moderating the compression is not as great as might be supposed, provided powerful ignition is installed.

Usual compression ratios are from 4.5 for hydroplanes to 5.6 for high-altitude land machines. Values of 6 or higher may be used for high-altitude work, but preignition will develop at full throttle near the ground.

Working fluid is not pure dry air,- specific heat is not constant, increase considerably at very high temperatures.

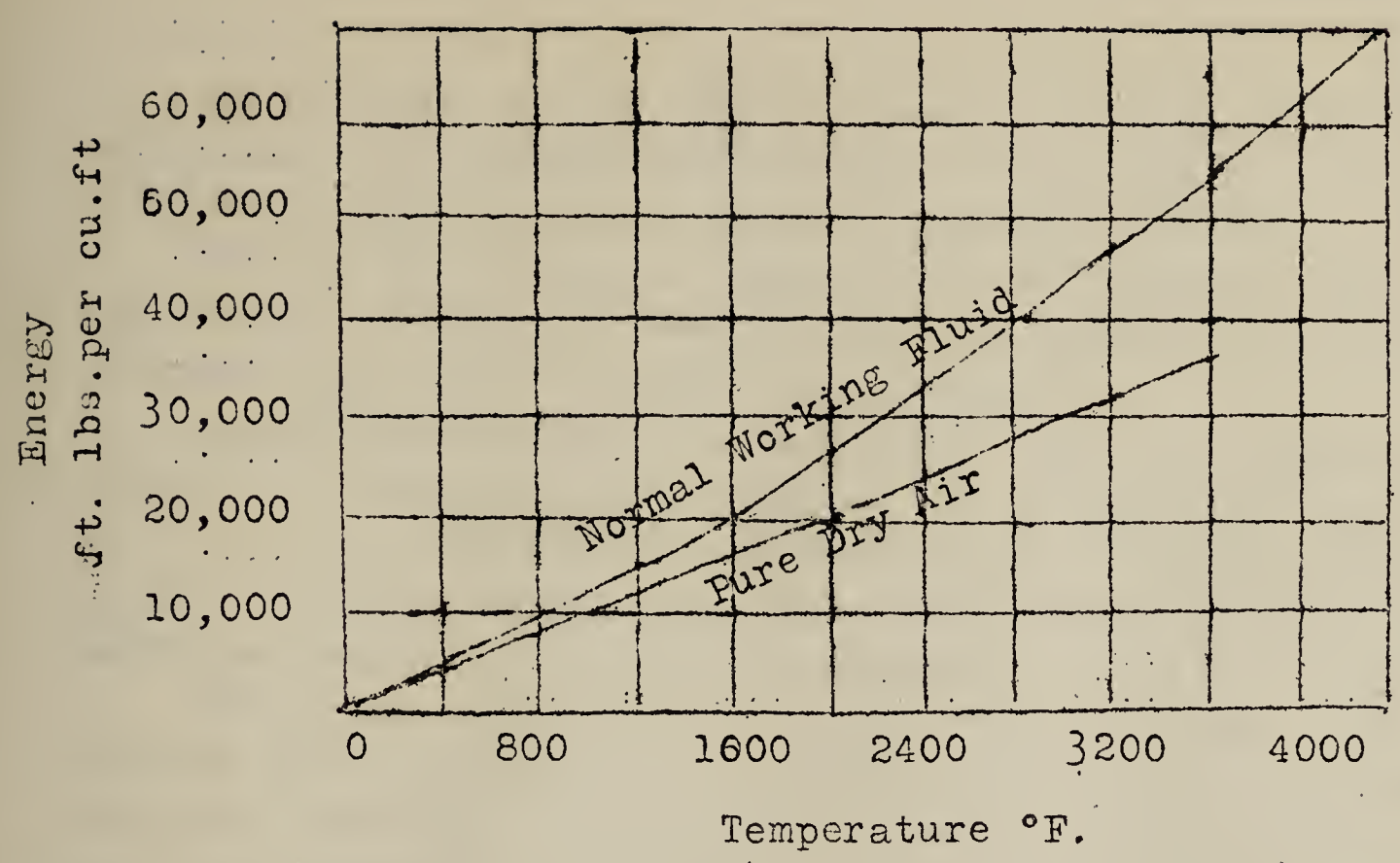


Diagram showing total amount of energy in working fluid at various temperatures. Recent research suggests that these values are too high; generally accepted, however, at present time.

If mixture ignites slowly, resulting flame takes comparatively long time to spread through the mixture. When mixture is completely inflamed, explosion takes place and maximum pressure is attained; time of complete inflamation is also that of maximum pressure. Time of complete combustion follows that of explosion and maximum pressure.

When a wave of compression is set-up, detonation is caused. The combustion will spread just as rapidly as the compression wave travels through the mixture. Speed is very considerable as compared with that of inflamation.

Two modes of combustion approached under certain conditions. Normal combustion is neither the one nor the other.

Experiments on rates of flame propagation in various mixtures under different degrees of compression have established the following facts:

The rate of flame propagation in explosive mixtures:

(1) is diminished as the proportion of inert gas present is increased;
(2) is dependent on the proportion of the active constituents;
(3) is diminished as the rarefaction of the mixture increases;
(4) is increased as the temperature of the mixture is raised;
(5) is much greater when the mixture is ignited at constant volume than when ignited at constant pressure.

As extremes of the proportions of mixture for possible explosions are approached, namely 22 to 1 and 8 to 1, the weakened effects of explosion are due to different causes.

Effect of increased temperature is to widen the range of dilution; Effect of increasing the compression pressure is to increase the rate of flame propagation.

Experiments of Dr. Neumann show that the velocity of flame travel is a maximum for a proportion of gasoline vapor and air of 1 to 12. The proportion for complete combustion is approximately 1 to 15; so, the maximum flame velocity occurs when the gasoline vapor is slightly in excess.

The flame velocity is much greater when the mixture is exploded under conditions of constant volume than when under constant pressure. Ignition at constant volume can only occur at the dead-center position, so that for maximum flame velocity it is important to ignite as near the dead center as possible, at the same time bearing in mind the necessary time interval elapsing between the point of ignition and the attainment of maximum pressure.

The larger the capacity of the combustion space, the longer the period elapsing between ignition of the charge at the attainment of maximum pressure. Also, the larger the explosion space, the slower is the rate of cooling.

The rapidity of the rise in pressure from the time of ignition is a measure of the explosibility of the mixture and the interval elapsing between the time of ignition and attainment of maximum pressure is termed "time of explosion".

The time of explosion for the same mixture can be considerably increased by igniting an appreciable mass of the charge at once.

The shape of the explosion chamber has a direct influence on the explosion period. The maximum pressure is not increased by rapid ignition.

Increasing the density of the mixture results in an almost proportional increase in the maximum explosion pressure. Also, if the density is increased, the rate of cooling is diminished.

There is a large variation in temperature at different points within the explosion space.

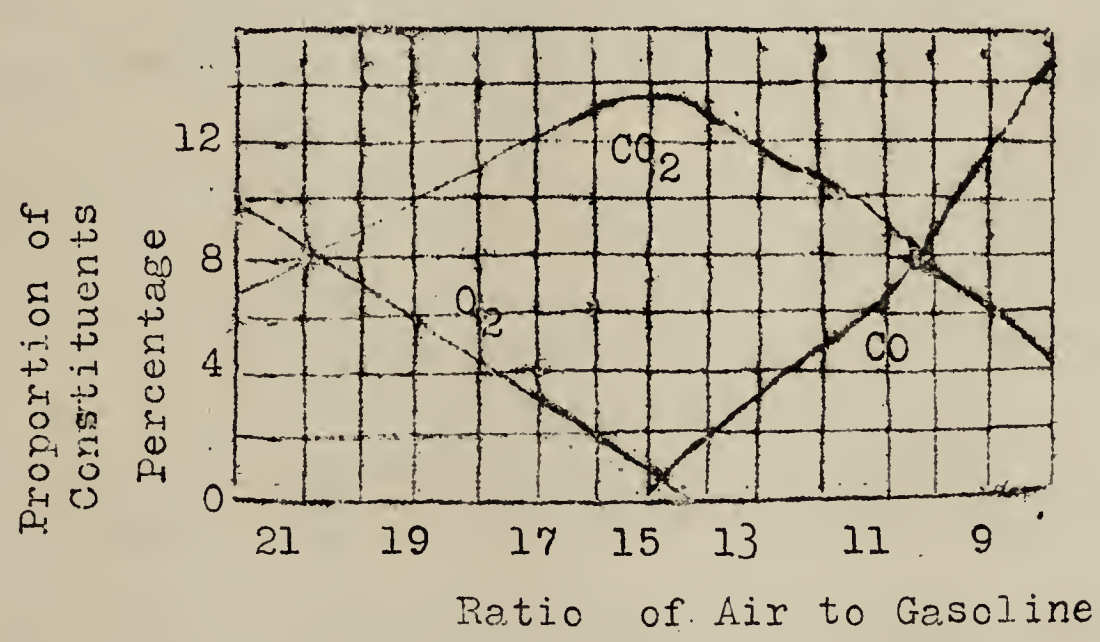
Measurements from indicator diagrams show that the rate of flame propagation varies from 35 to 100 feet per second.

At high speeds the explosion interval, the interval between the moment of the spark occurring and the attainment of maximum pressure, is about $\frac{1}{250}$ of a second; while the complete combustion period is about $\frac{1}{150}$ of a second. Here, turbulence is important; for it greatly increases the rate of flame propagation.

The volume of the products of combustion is not the same as the volume of the original explosive mixture. The final volume will be greater. For weak mixtures the increase in volume is about 5%. While for a perfect mixture the increase in volume is between 5% and 6%. For rich mixtures the increase in volume may reach 15%.

Composition of Exhaust Gases. Predictions are borne-out in practice; the exhaust consists of carbon dioxide and oxygen for weak mixtures; and of carbon dioxide and carbon monoxide for rich mixtures.

Results of a large number of tests and analyses of exhaust gases in a four-cylinder engine by Dr. Watson.



Note that the percentage of oxygen present becomes greater as the mixture gets weaker, that as the mixture becomes succeedingly richer the percentage of carbon monoxide becomes greater;

and that for weak mixtures no carbon monoxide is present, while for richer mixtures no oxygen is present. That both CO and O_2 do not occur together has been established.

Occasionally the ratio of the proportions of air and gasoline in the mixture have to be determined from such an analysis; and frequently this method is the only one used for finding mixture strengths.

The mixture which gives the nearest resemblance to perfect combustion is about 14.75 parts of air to 1 part of gasoline.

The whole of the calorific heat of the fuel is not liberated during the explosion process. As the mixture becomes richer, the percentage of the heat of the fuel liberated becomes smaller.

The actual temperatures and pressures obtained in explosion engines, are very much lower than the theoretical ones calculated. The explosion temperatures are often found to be only about one-half of the theoretical, and the pressures only about 50% or 60% of the pressures that ought to be attained. Also, only about one half of the heat energy of the fuel is apparently developed in the explosion.

Several theories advanced in explanation of the apparent loss. None account for the entire discrepancy, but each gives explanation for some of the apparent loss.

Dissociation of the gases appears to consist of a suppression of heat at the maximum temperature of explosion, and an evolution of heat during the expansion stroke. It plays an important part in combustion, and temperatures are thereby limited. Dissociation is only one of the contributing causes.

Cylinder wall action, a partial explanation of heat suppressio The charge loses heat to the walls both by radiation and by conduction.

Three important factors concerned in wall action,- Radiation Conduction, and Turbulence.

Radiation is almost of same importance as conduction. Heat flow and temperatures thereby involved are of great importance to the designer, in connection with temperature stresses and distortions, allowance for effects of temperature changes, density, composition of the mixture and other factors.

The heat loss by radiation from the exploding and expanding gases to the cylinder walls depends on,- the mean temperature and density of the gases, the area and shape of the cylinder walls and the state of their surface. Probable that heat loss is proportional to the surface area, and is independent of time or engine speed; also, that part of radiation loss during combustion is proportional to the volume and not the surface. Radiation from hot gases is rendered more complex because gas is transparent to its own radiation to an appreciable extent,- radiation dependent on thickness of its mass.

Effect of enrichment of the mixture is to increase amount of heat radiated.

Effect of pressure or radiation cannot be exactly stated,- radiation varies about as the density raised to the 0.9 powers. As density of charge is a maximum near end of compression stroke, radiation will here be the greatest.

Quantity of heat lost to walls depends also on state of their surface, - whether polished or coated with heat absorbent. Results of experiments show that heat efficiency and mean effective pressure will be appreciably greater with a polished combustion chamber than with a carbon-coated surface.

Should be recognized that it does not seem possible to determine the effect of the various factors with sufficient accuracy to enable any quantitative inference to be drawn as to radiation from flame experiments in the case of explosion engines.

Effect of Turbulence on rate of ignition and heat flow. High velocity of gases produces eddying effects, and this eddying motion persists throughout compression and still exists when ignition takes place. Illustration and discussion.

Suction turbulence of vital importance in producing high rates of flame travel necessary in explosion engines.

Not only is maximum pressure increased by turbulence, but the period of time of attainment is reduced. Actual ignition velocities and pressures attained in gasoline engines are only possible as a result of turbulence.

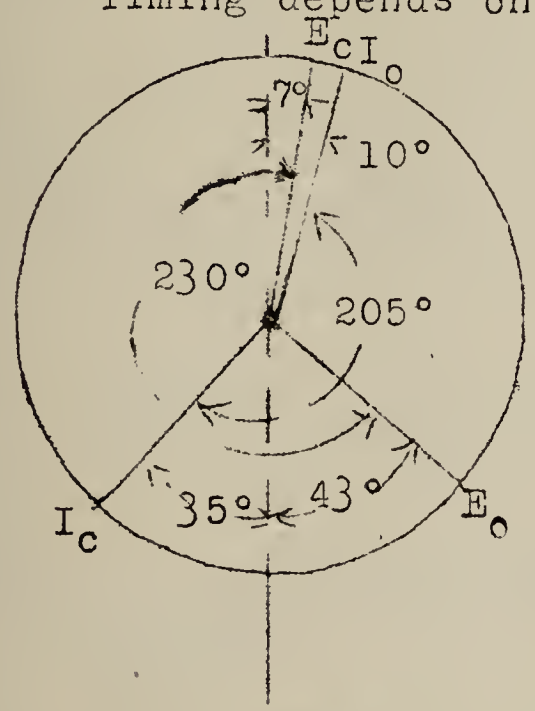
Design of combustion chamber and valve passages have importan bearing on rates of ignition corresponding with different modes and degrees of turbulence effect.

Experiments on sleeve-type and T-head motors, - more rapid fre rate of ignition, and maximum pressure attained sooner with engine from valve pockets and with central ignition.

Valve Timing

Correct timing of valves of great importance, since valve action influences to marked degree the performance of the engine. Too early or too late opening or closing of either the inlet or exhaust valve effects the H.P. output and thermal efficiency.

Timing depends on several conditions, and general design of the engine. Diagram of fairly average valve timing for high speed motor.



Inlet Valve. When piston has reached end of suction stroke, pressure in cylinder is still below atmospheric, so inward flow will continue after piston has advanced upward. Therefore, inlet valve is closed late,- 35° - 50°. Inlet valve should remain open as long as gas will flow into the cylinder, in order to obtain full charge. Time of inlet valve closing depends on maximum speed of engine, number of cylinders, size and location of valves.

Time of opening may be from 0° to 15° late.

Desirable features to be gained by the inlet valves.

1. To induce the maximum possible weight of charge into the cylinder.
2. To expend the least possible energy in the process.
3. To produce the maximum of turbulence during the period of entry.

Volumetric efficiency and power-output depend largely on inlet valve timing. Negative work during suction stroke must not be unnecessarily high. Consideration must be given to how many cylinders are drawing supply from one source and how far maximum power should be subordinated to economy at reduced loads.

Charge induced into cylinder by suction action of piston; useless to open valve before dead-center, it should remain open after passing dead-center to give mixture opportunity to fill the cylinder.

Time of opening and closing varies with designed speed of engine.

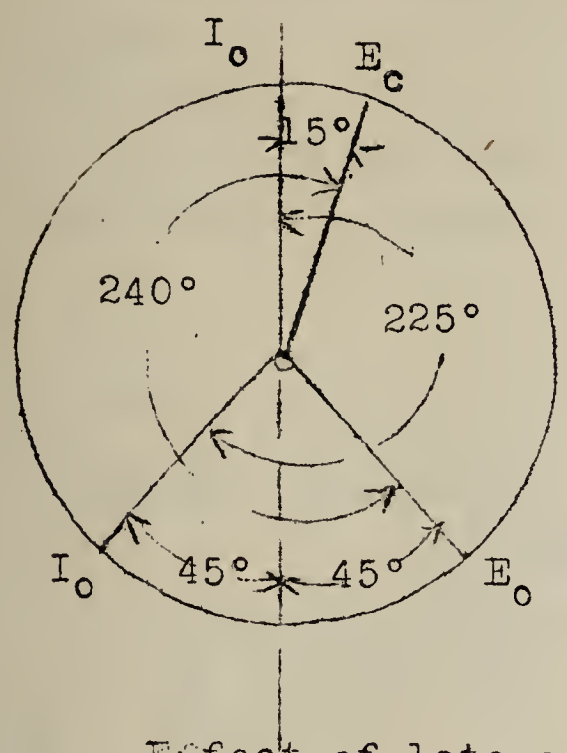


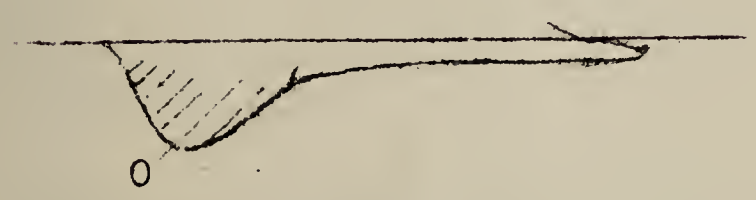
Diagram for very high speed engine.

In some engines, inlet valve opens at very small angles and in a few cases before dead center.

Time of opening of inlet valve bears relation to time of closing exhaust valve. Practice differs.

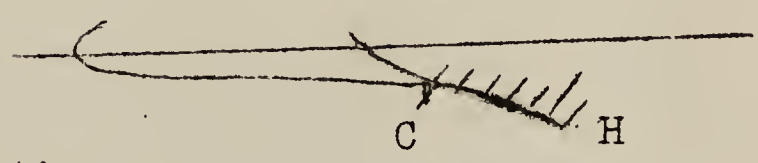
Inlet valve should not be opened until pressure in cylinder has fallen to atmospheric.

Effect of late opening is shown in diagram. Suction increases rapidly until time of valve opening, O.



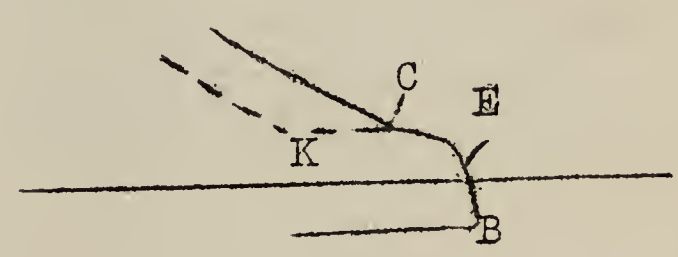
Time of closing the inlet valve the most important factor; has marked effect on maximum power and maximum attainable speed.

If closed too early, suction increases rapidly from time of closing, line C H.



Shaded portions represent power expended in unnecessary suction. Real harm done, incomplete charge.

If closed too late part of charge will be thrown back into manifold, line B C. C, time of closing. Line K C shows effect of still later closing at K.

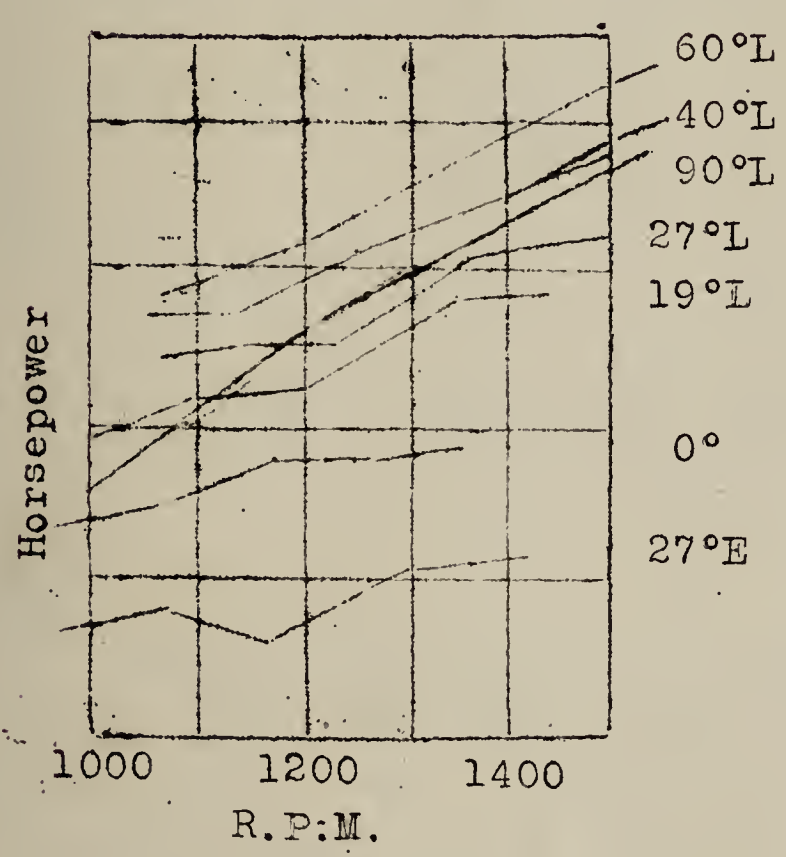


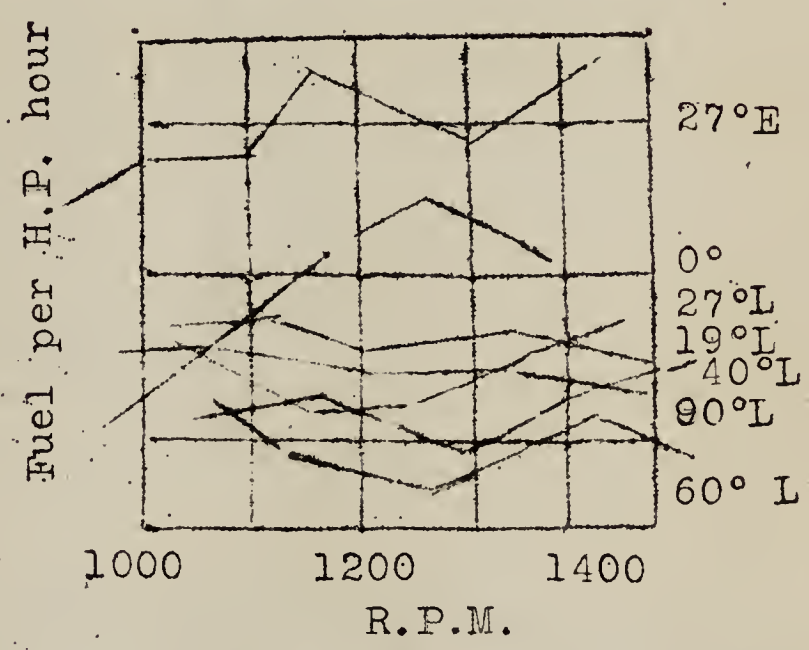
Important to close inlet valve at just the right moment, otherwise loss occurs.

In general, the higher the designed engine speed the later the time of closing the inlet valve and the earlier the opening; also, the later the closing of exhaust valve.

If valve timing is one for giving maximum power at high speeds, the engine will not run as well or be as flexible at low speeds, - due to difference of inertia of gases.

Results of tests on four cylinder engine with seven different inlet-valve closings.

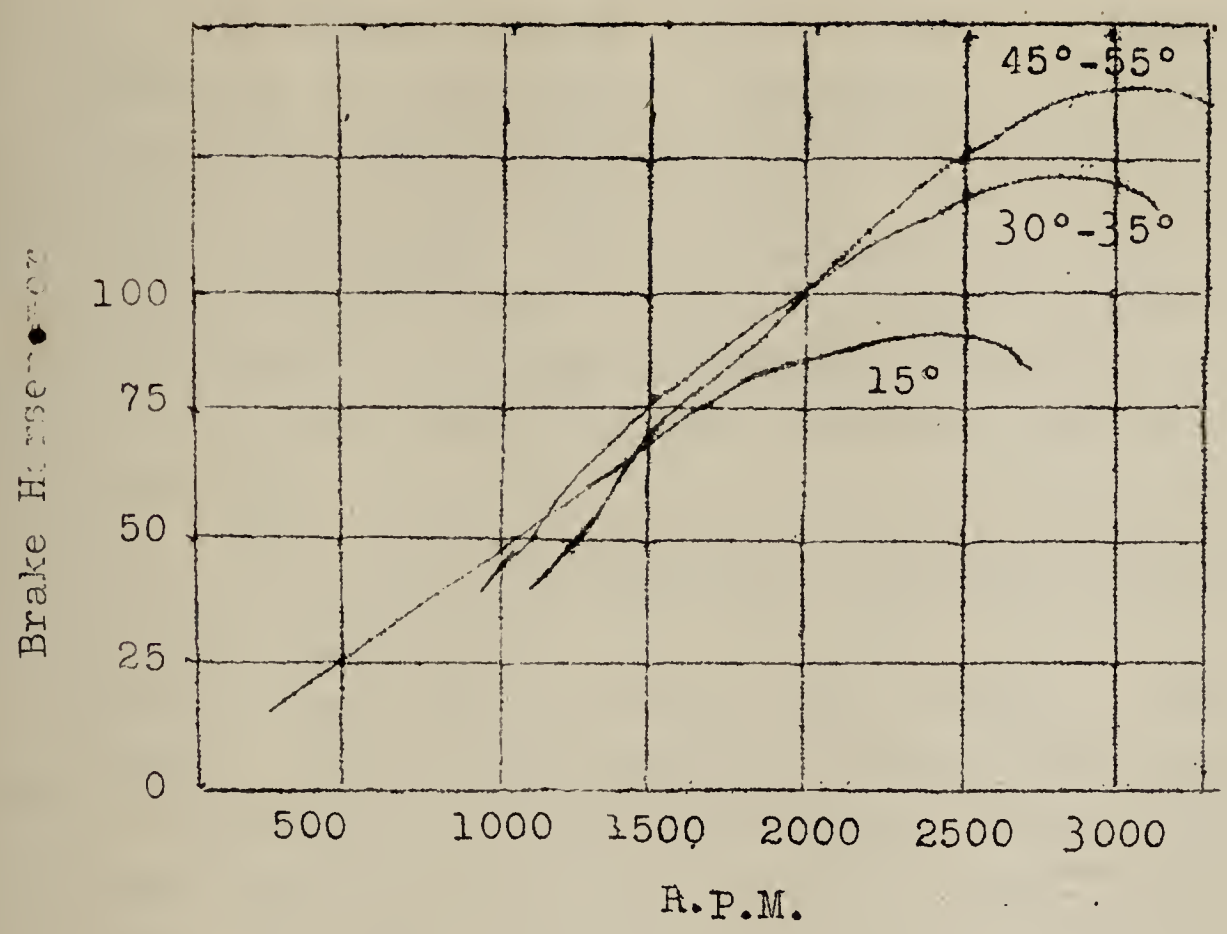




Maximum horsepower was for valve closing 60° late. The higher the speed, the later the inlet valve should close. Effect on thermal efficiency similar to that on horsepower.

Fuel consumption on same tests show the most economical time for closing the inlet valve was 60° late.

Test not carried to a high speed. Although 60° late timing was best for the higher speeds, it was not the best for lower speeds. Compromise would be made if a flexible engine is desired.



Results of tests on a modern high speed gasoline engine for three different inlet valve closings, support the previous general conclusions; but it should be noticed that up to 2100 r.p.m. (a much higher speed than the previous series) the 30° - 35° timing seems the best.

Exhaust Valve. Pressure in cylinder during exhaust stroke should be as near atmospheric as possible; necessary to open exhaust valve in advance of lower dead center to give burned gas free opening for escape when piston starts on the exhaust stroke.

Number of degrees of angle depends on, - ratio of area of valve opening to cylinder displacement, speed of engine, design of exhaust manifold, character of exhaust cam.

Although some force of expansion is wasted by the early opening, the loss is more than compensated-for by freedom of exit. Too early an opening would give uncompensated loss. Time of opening is usually about 45° early.

If exhaust valve closes at end of exhaust stroke where pressure is still above atmospheric, it will entrap in the clearance space unwarranted amount of waste gases which will vitiate the new charge unduly. Gas will flow-out after dead center is passed, due to its inertia. The valve should not be closed while there is still chance of burned gases escaping. The angle of closing is usually from 5° to 15° late.

In any case, the back pressure should be kept as low as possible.

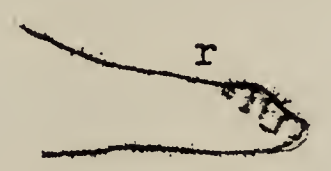
The problem is that of getting rid of the exhaust gases with the least possible back pressure and the least distress to the valve gear.

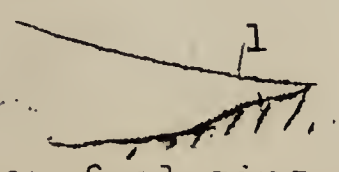
Two factors must be taken into account. The valve must be opened sufficiently early to permit the exhaust pressure falling to atmospheric before the exhaust stroke begins; it must be held open sufficiently late to permit the residual exhaust gas escaping. No exact figures can be given because of many variable factors, such as mean gas velocity through valve opening, and rate of acceleration of the valve.

Ricardo's opinion for best setting of the exhaust valve.- It should already have traveled through 50% of its total lift when the piston is at the lower dead center, and should still be 5% open when the piston is at the top dead center. The actual points at which the valve leaves its seat and returns to its seat are of less significance.

The exhaust valve should be so timed that the loss resulting from an early opening, plus the loss arising from the back pressure will be a minimum.

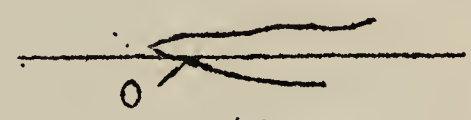
If valve is opened too early, the end of expansion curve will fall to atmospheric before end of stroke; when valve opens at "r", the shaded area represents the power lost. If valve is opened too late there will be a loss due to rise of back pressure; when valve opens at "l", the shaded area represents the power lost.





Time of closing exhaust valve. If it is closed too early, an excess of burned gas will remain in the cylinder. If it is

kept open too long; burned gases will re-enter the cylinder with the fresh charge. As the exhaust line is always above the atmospheric pressure, there is never danger of gases being sucked-back into the cylinder during the exhaust stroke, nor until the piston on its suction stroke passes the point, O, where the pressure is equal to atmospheric.



When the exhaust valve is opened, the pressure in the cylinder is considerable; burned gas will escape at high velocity. If the valve is kept open at end of stroke, the inertia of this gas as it passes into the exhaust pipe will draw after it some of the burned gas. This method of scavenging often realized by closing exhaust valve late.

Inertia of exhaust gases sometimes used to assist the induction of fresh charge; inlet valve is opened before exhaust valve closes.

Valves should be timed for best working of engine at speed at which it is mostly intended to run. Various timings, based on previous results are tried, in order to find timing that gives best results.

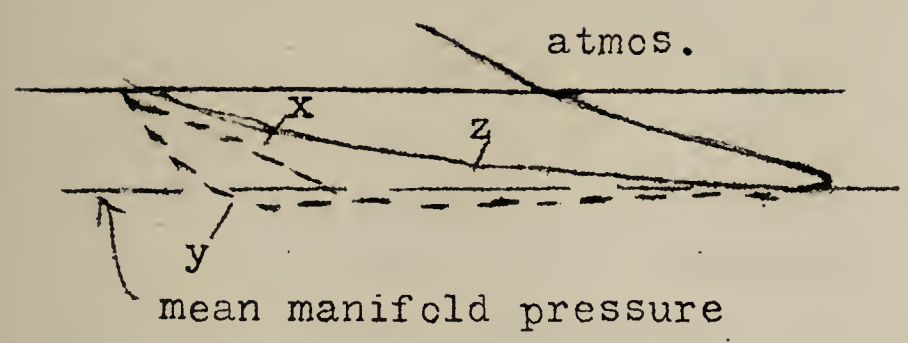
<u>Fuel Economy</u>. If better fuel economy is desired at sacrifice of maximum horsepower, the period of opening of inlet valve would be extended. Opened considerably earlier to reduce pumping loss; and closed little later to give longer time for filling the cylinder.

<u>Throttling</u>. Aim is not to obtain full charge, but to maintain turbulence and reduce pumping losses as far as possible.

To maintain turbulence we can rely only on the use of smallest possible valves, on keeping the orifice coefficient as high as possible, and an unobstructed entry to the cylinder after passing the valve.

Another point of importance is position of throttle.

If throttle is close to valve port there is small capacity between it and valve; so, during idle strokes, capacity will fill-up to atmospheric. In this case inlet valve should be opened early. See line "Z" in diagram.



If there is considerable capacity between throttle and valve, at beginning of suction stroke, pressure in cylinder will be atmospheric while that in port will be below atmospheric; so, when inlet valve is opened pressure in cylinder will be reduced, and there will be unnecessary negative work, as shown by line "y". In this case it would be preferable not to open the inlet valve until gas in cylinder has expanded to pressure corresponding to that in port, - shown by line "x".

Not practicable to obtain a valve timing which will be ideal for all conditions of load. The best that can be done is to arrange a timing to suit the load at which the engine will run for the majority of the time.

Multiple Cylinder Engines. When several cylinders draw their supply of mixture from a single source, the timing of inlet valve is much more difficult. Valve timing and distribution cannot be treated separately; they are closely interdependent. Together, form intensely complex problem, probably least understood of all problems connected with the modern internal combustion engine.

Effect of Valve Timing on Efficiency. For same compression ratio, timing of valves has noticeable effect on thermal efficiency.

Approximate correction for effect of delayed inlet-valve closing on air-standard efficiency.

$$E = 1 - \left(\frac{2}{(1 + \cos\alpha)(\text{comp.ratio})}\right)^{n-1}$$

where α is the delayed angles.

Formula is suitable only when α is small.

Heat loss due to too early opening of exhaust valve; varies from 3% to 6% of total heat of fuel in high speed engines. Cannot be much reduced without effecting scavenging and volumetric efficiency.

Adjustments and Tappet Clearance. Valve adjustments should be made when engine is hot. Valve clearance should be as small as possible consistant with assurance of valves seating. Too large a clearance produces noise and possibility of breakage of valve-gear parts.

Thermodynamics

From the thermodynamic point of view, the fundamental properties which define the physical state of a gas are its pressure, temperature and volume; with a knowledge of these three factors, the other properties can be deduced.

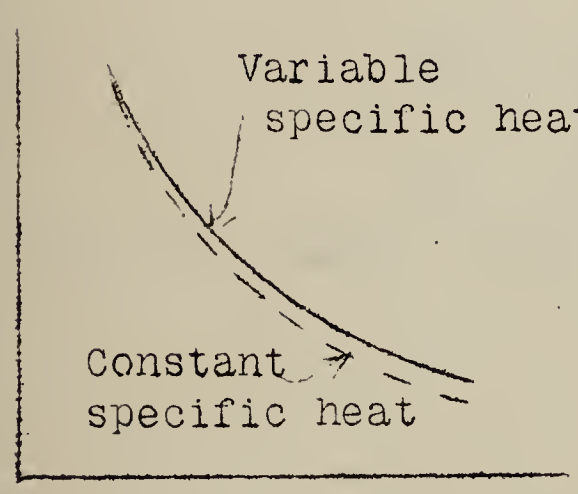
With gases which deviate from the ideal state as expressed by Boyle's and Charles's law, the specific heats are not constant. Their ratio "n" diminishes as temperatures are raised.

Variation of "n" with Temperature.

	100° C	600° C	1100° C
Air	1.404	1.38	1.345
Steam	1.30	1.29	1.24
CO_2	1.26	1.20	1.18

Most recent experiments are not in exact accordance with these values.

Effect of variability of specific heats on the adiabatic curve is to make it less steep than when PV^n = constant is followed.



An empirical method for obtaining the adiabatic lines in case of variable specific heats consists of splitting-up the volume line into a number of ordianates, and finding values of "n" for these corresponding temperature ranges. With this method several trials would have to be made before the correct value of "n" could be found.

Curve for total energy in the mixture at different temperatures is useful in making calculations of temperature and pressure, and is a fairly simple means of taking account of variation of specific heats with temperatures.

Total heat of a gas is represented by sum of its internal energy and work done in expansion; also, total heat is independent of form of path from one state to the other.

Experiments conclusively prove that no gas obeys either Boyle's or Charles's law exactly.

In calc ulations on the internal combustion engines, we have to deal with mixtures of gases instead of a single gas; and it is necessary to know the value of the gas constant, R, in equation $PV = RT$, in such cases.

For mixture of two gases,

$$R = \frac{M_1R_1 + M_2R_2}{M_1 + M_2}$$

where M_1 and M_2 are the masses of the gases at same volume.

For a mixture of several gases

$$R = \frac{\sum MR}{\sum M}$$

Specific heats of gaseous mixtures are easily found when the percentage composition of the mixture and the specific heats of the constituents are known

$$C_p = \frac{\sum W C_p}{\sum W} \quad \text{and}$$

$$C_v = \frac{\sum W C_v}{\sum W}$$

where W is the weight of the constituent gases.

Problem

Airplane Engine,- 5" X 6-1/2"

Connecting Rod length = 11-1/2".

Clearance = 23 %.

Compression Ratio = 5.35.

Rev. per min. for maximum brake horse-power = 2000.

Take atmospheric pressure as 14.5 pounds, and temperature of atmosphere as 50° F.

Calculate the pressure and temperature at the five important points of the cycle, and draw the ideal indicator diagram. Find the ideal efficiency.

Produce a probable actual indicator diagram for the engine, finding pressures and temperatures at the five important points of the cycle. What is the actual efficiency? Find the mean effective pressure and the indicated horse-power.

Conditions in Engine Cylinder

Important to know pressures and temperatures existing in engine cylinder. Difficult to find probable pressures and temperatures at different parts of cycle; calculation involves knowledge of variation in specific heats of the gases, nature and amount of cooling losses, volumetric efficiency and other indeterminate factors.

After obtaining an ideal indicator diagram for purpose of comparison of performance of an engine, a probable diagram for the given engine should be produced.

Modified Ideal Diagram. Correction for effect of varying specific heats; must find value of "n", (PV^n = constant), for the expansion curve. Pressures and temperatures for the suction and compression strokes remain unchanged.

For condition at end of explosion proceed as follows:

Work done on gas during compression would be

Heat energy added = working capacity of cylinder

$(T_3 - T_2)$ X specific heat at constant volume.

Adding this heat energy to assumed heat supply during combustion gives internal energy at end of explosion. Refer to curve of internal energy and read-off temperature corresponding, and calculate pressure. Specific heat varies during expansion stroke. Process of trial and error to find mean value of "n" that satisfies amount of internal energy. With this value of "n", expansion line can be drawn and pressure and temperature at release calculated; values will be higher than ideal.

$$\text{Efficiency} = \frac{\text{Available energy}}{\text{Total energy}}.$$

Available energy is available work in expansion minus work done by compression; total energy is energy (assumed) supplied during combustion.

Efficiency is lower than that under assumption of constant

specific heat. It is, therefore, the efficiency of the ideal engine with which the real performance of actual engine should be compared; for it represents the maximum efficiency that an actual engine can possibly attain, taking into account the properties of the working gases.

Other Methods. Account taken of cooling effect and increase of volume of gases after combustion.
The problem of estimating actual temperatures and pressures is complicated, as actual state of affairs existing is only imperfectly known. Knowledge of duration of period of combustion, nature and magnitude of wall action, radiation and other factors is necessary before exact temperatures and pressures can be calculated.

Actual Diagram.

Suction Temperature and Effect of Residual Exhaust Gases.
To estimate more exactly the temperature of suction, it is necessary to consider amount and temperature of exhaust gases in cylinder at end of suction stroke.
Effect is to increase suction temperature.
Volume of fresh charge is

$$(\text{comp. ratio} - 1)\, v$$

where v is the clearance volume.
If density and specific heats remain constant,

$$\frac{(\text{comp. ratio} - 1)v}{t} + \frac{v}{T_e} = \frac{(\text{comp. ratio})v}{T_s}$$

where t is temperature of atmosphere and T_e and T_s temperature of exhaust and suction.

$$\therefore\ T_s = \frac{(\text{comp. ratio})\ t\ T_e}{(\text{comp. ratio} - 1)\ T_e + t}$$

Compression Line. Not exactly adiabatic. Heat energy does not remain unaltered; pressure and temperature at end of compression less than for exact adiabatic.

Value of "n" generally between 1.2 and 1.4.
1.35 not unusual.

Maximum Pressure. Rise of pressure after ignition much less than predicted under assumption that all heat of combustion is used-up in raising temperature and pressure. Factors contributing to pressure suppression,- dissociation, wall action, late combustion, increasing specific heat of mixture; explosion not instantaneous, occupies period of timedependent on proportion of mixture, time of ignition, speed of engine, size of cylinder. compression pressure and other factors,- heat not given to charge at strictly constant volume.

Expansion Line. Combustion continues after point of maximum pressure is attained; addition of heat to expanding gases, opposing effect of absorption of heat by walls. Actual line differs from true adiabatic.
Value of "n" is generally between 1.15 and 1.35.
With weak mixtures, expansion line does not follow any definite law.

Exhaust Line. Exhaust valve is opened before end of expansion stroke; end of diagram rounded-off. Exhaust valve held open long period; heat rejection not same as in ideal case.

Actual Efficiency

Ideal diagram obtained under assumption that are not realized in actual engine.

Air standard efficiency depends solely on compression ratio.

$$\text{Air Standard Efficiency.} \quad E = 1 - \frac{1}{\text{comp.ratio}^{1-n}}$$

Compression Ratio	Efficiency E	Compression Ratio	Efficiency E
3.25	0.382	4.75	0.481
3.50	0.400	5.00	0.492
3.75	0.416	5.25	0.501
4.00	0.445	5.50	0.510
4.25	0.458	5.75	0.519
4.50	0.470	6.00	0.548

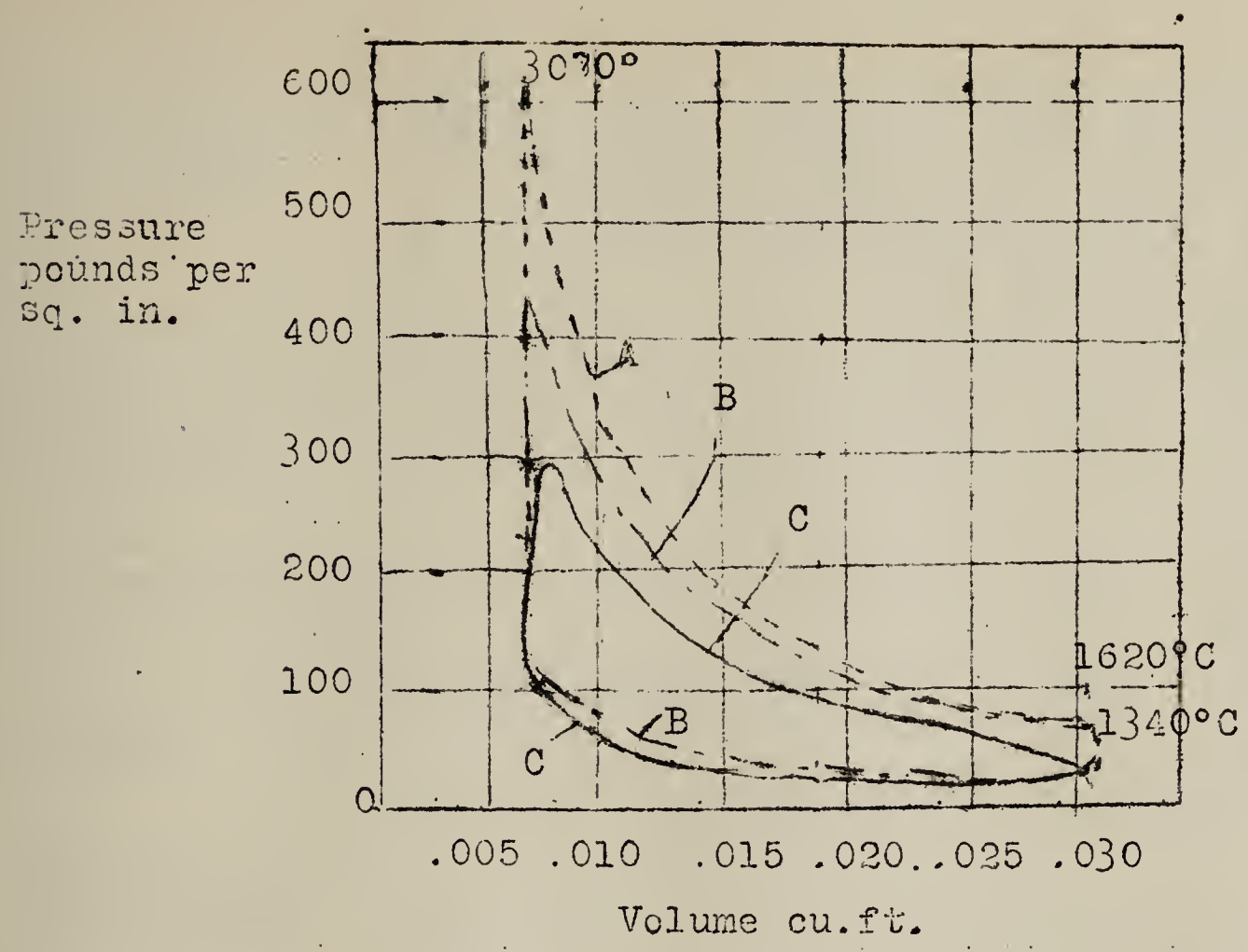
Work of Dr. Watson on engine of working capacity of 0.025 cu.ft., compression ratio, 4.7, fuel mixture 14.1 to 1, heat supplied 1150 ft. pounds.

A. Indicator diagram for ideal engine working on air cycle.

B. Indicator diagram for ideal engine working on variable specificheats.

C. Indicator diagram of actual gasoline engine.

The Table accompanies diagrams A, B, C.



Mixture, 14.1 to 1. Heat supply 1150 ft.lbs.	Efficiency	Max. Pres. lbs.-sq.in.	Mean Press. lbs.-sq.in.	Heat converted into useful work per cycle. ft.lbs.	Heat rejected per cycle ft.lbs.
A. Ideal engine, air cycle	.463	625	153	532	618
B. Ideal engine, variable specific heat cycle	.356	448	118	410	740
C. Actual gasoline engine	.248	290	84	285	865

Other tables for a weaker and a richer mixture.

Mixture 17.4-1. Heat Supply 962 ft.lbs.	Efficiency	Max. Press. lbs.-sq.in.	Mean Press. lbs.-sq.ins.	Heat converted into useful work per cycle ft. lbs.	Heat rejected per cycle ft. lbs.
Ideal engine air cycle	.463	550	128	445	517
Ideal engine variable specific heat cycle	.367	400	102	352	610
Actual gasoline engine	.275	207	77	265	697
Mixture, 10.1-1 Heat Supply 1027 ft.lbs.					
Ideal engine, air cycle	.463	565	137	475	552
Ideal engine, variable specific heat cycle	.381	440	113	390	637
Actual gasoline engine	.289	300	86	297	730

Advantages in using Variable Specific Heat Standard instead of ideal Air Standard as a comparison of efficiency of actual engine. Because, the variable specific heat standard represent the ideal practically attainable standard when using same working mixture as that of the actual engine. Its efficiency is lower than the air standard, and appreciably higher than that of the actual engine; owing to the fact that no cylinder heat losses are pre-supposed in the ideal case.

The actual thermal or indicated efficiency is the ratio of the useful work obtained at the piston to the heat supplied during the cycle.

Relative Efficiency of Actual Engine and Ideal

Mixture: Air to gas ratio, by weight	Efficiency / Air Standard Effic.	Efficiency / Variable Spec. Heat Stand. Effic.
10.1	.624	.759
14.1	.535	.696
17.4	.594	.750

Variation of Efficiency with Compression Ratio.
Relative Efficiencies

Comp. Ratio	Comp. Pressure in lbs. per sq. in. above atmospheric at 1000 r.p.m.	Actual Effic. of Engine	Air Standard Effic.	Ideal Var. Spec. Heat Effic.	Effic. / Air. Standard Effic.	Effic. / Ideal Var. Spec. Heat Effic.
4.71	86	.275	0.463	.367	0.594	.749
4.35	77	.272	0.450	.354	0.604	.769
3.92	68	.262	0.425	.337	0.615	.780

The table shows results for same strength of mixture and same speed of engine.

Actual or measured efficiency depends on both the mixture strength and engine speed.

Efficiency of actual engine only about 76% of specific heat standard efficiency, and only about 60% of air standard efficiency.

Opportunity for obtaining larger theoretical improvement through use of working substance that more nearly approaches the air standard; also possible use of higher compressions. Greatest possible gain through elimination of all losses of heat to cooling water and to the exhaust is comparatively small, - about 20%; not open to much improvement.

Indicated Thermal Efficiency.

$$\text{Thermal Efficiency} = \frac{\text{Actual work done}}{\text{Heat supplied in fuel}}$$

If work done is the indicated horsepower, the efficiency is called "Indicated Thermal Efficiency", if work done is at crank shaft, the efficiency is known as "Brake Thermal Efficiency".

Ratio of the two efficiencies represents the Mechanical Efficiency - varies from 80% to 90% depending on losses in friction, unbalance, etc.

$$\text{Indicated Thermal Efficiency} = \frac{\text{Indicated H.P.} \times 33{,}000 \times 60}{W \times Q \times 778}$$

where W is the weight in pound of the fuel used per hour, and Q is the calorific value of fuel in B.T.U. per pound.

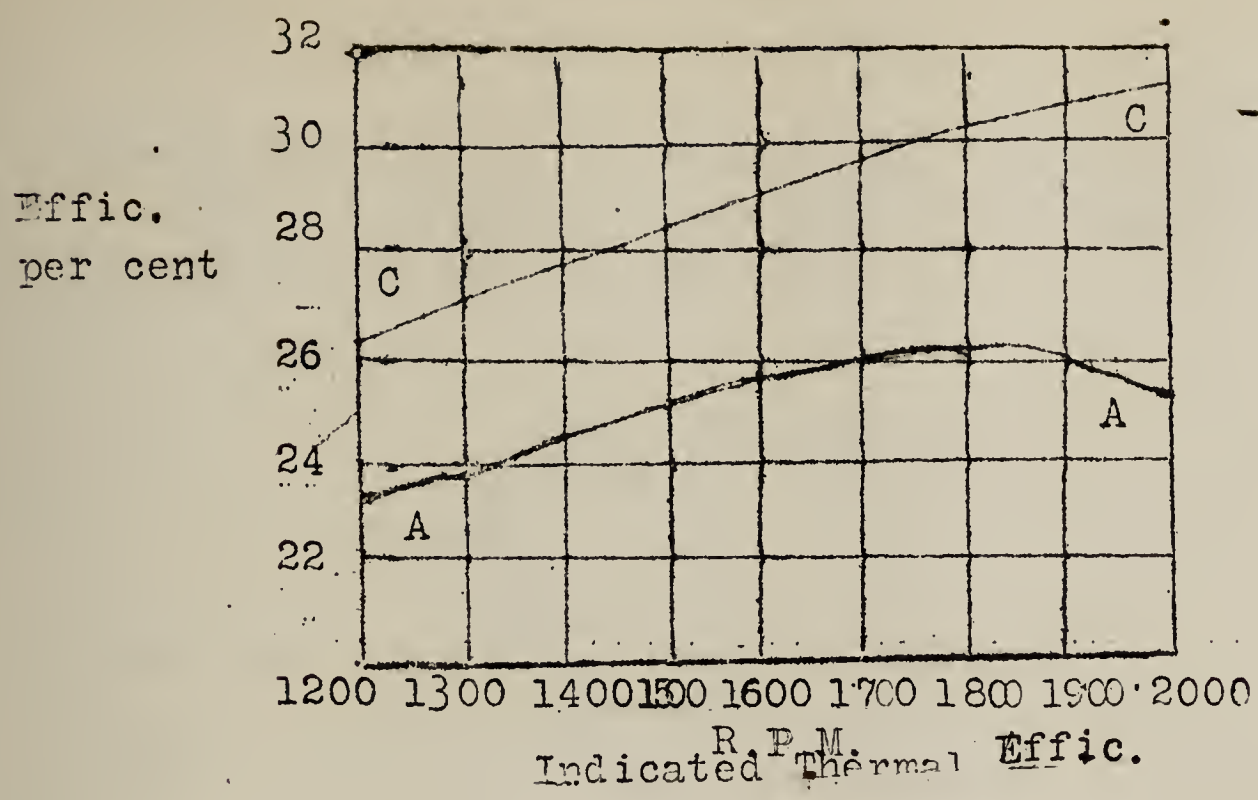
Brake Thermal Efficiency = Indicated Thermal Efficiency x Mechanical Efficiency.

Thermal efficiency increases as speed increases; fuel consumption diminishes as speed increases, up to a limiting speed.

Relation between thermal efficiency and fuel consumption.

Indicated thermal efficiency = 48.7 - 43.2 G where G is weight of gasoline in pounds per indicated horsepower per hour.

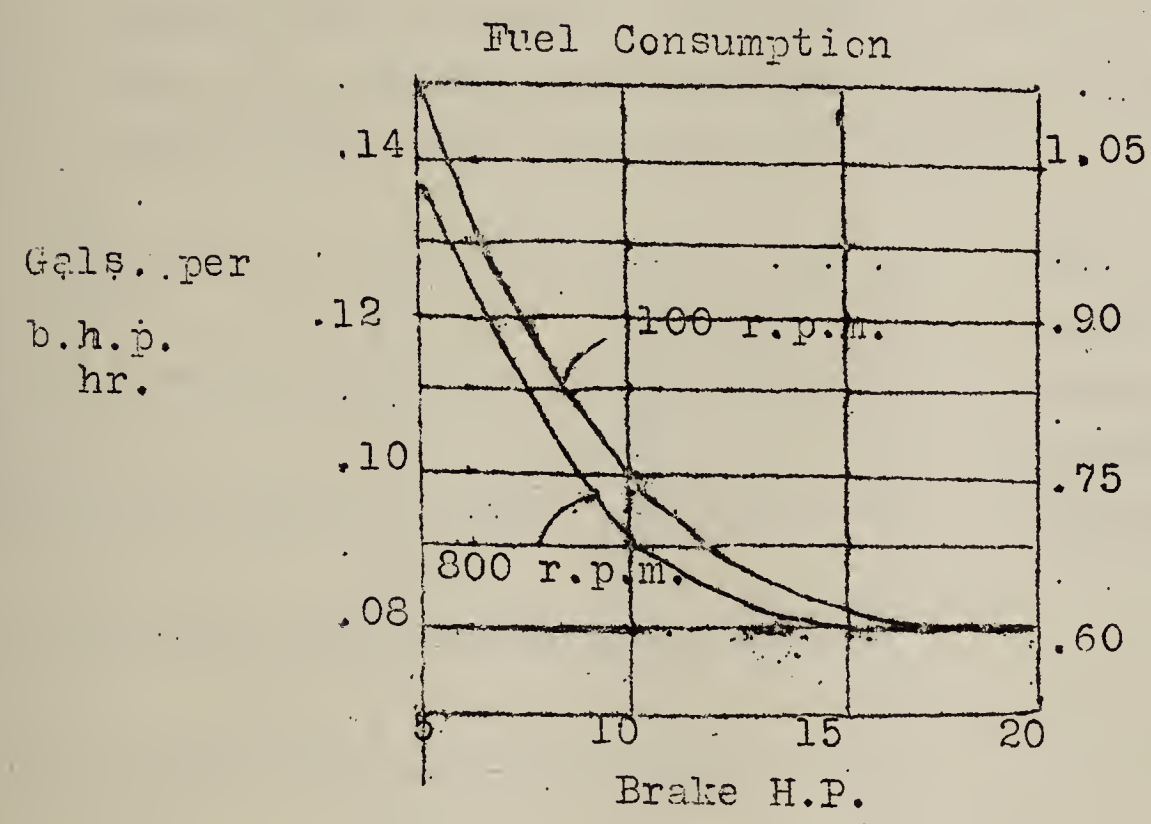
Results of tests on small four-cylinder auto-engine - L-head type, 2-3/4" bore x 4" stroke, compression ratio 4.45. Engine C was engine A with new design of cylinder head giving greater turbulence and higher compression, and with aluminum pistons in place of cast iron.



Relation between Fuel Consumption and Thermal Efficiency

R.P.M.	Engine	1,200	1,400	1,600	1,800	2,000
Indicated thermal efficiency	A	23.2	24.2	25.4	26.1	25.2
	C	26.3	27.7	29.1	30.1	31.3
B.H.P.	A	12.6	14.6	16.4	17.75	18.6
	C	13.6	15.8	18.0	20.2	22.2
Fuel (lbs. per I.H.P.-hr.	A	0.590	0.565	0.539	0.525	0.542
	C	0.521	0.494	0.470	0.454	0.437
Fuel (lbs. per B.H.P.-hr.)	A	0.691	0.667	0.647	0.646	0.690
	C	0.559	0.559	0.535	0.524	0.510
Thermal Ratio Effic. Fuel per I.H.P.-hr.	A	39.3	42.8	47.2	49.6	46.5
	C	50.4	56.1	62.0	66.4	71.6

Thermal Efficiency at Light Loads. Engine which gives maximum thermal efficiency a t about full load will give poor fuel economy at light loads.



Curve shows that at one-half load the fuel consumption is about 25% greater than at full load, and that at one-third load the fuel consumption is about 60% greater than at three-fourths to full load.

Factors Influencing Thermal Efficiency

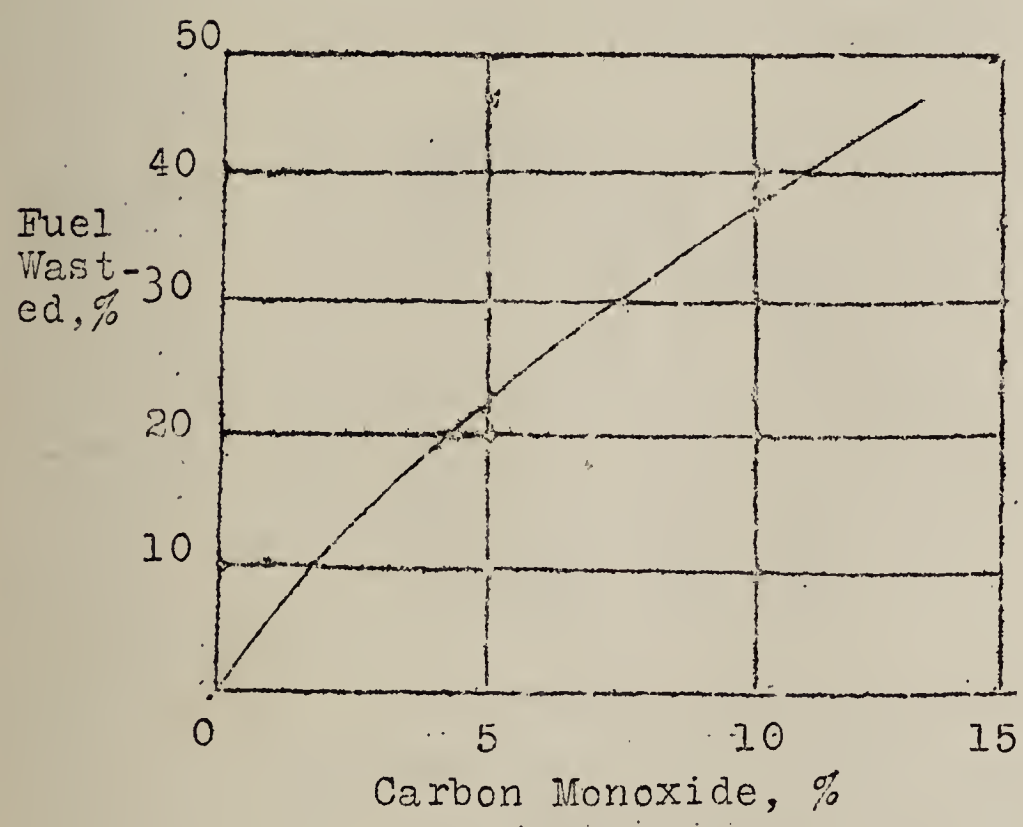
Thermal Efficiency of internal combustion engine depends on, -

1. Nature of fuel used
2. Proportion of air to fuel
3. Compression pressure
4. Size of cylinder and shape of combustion chamber.
5. Rationof stroke to bore.
6. Speed of engine
7. Amount of fresh charge admitted, throttle opening.
8. General design, valve openings, valve timing, cooling arrangements, etc.

Composition and Proportion of Charge.

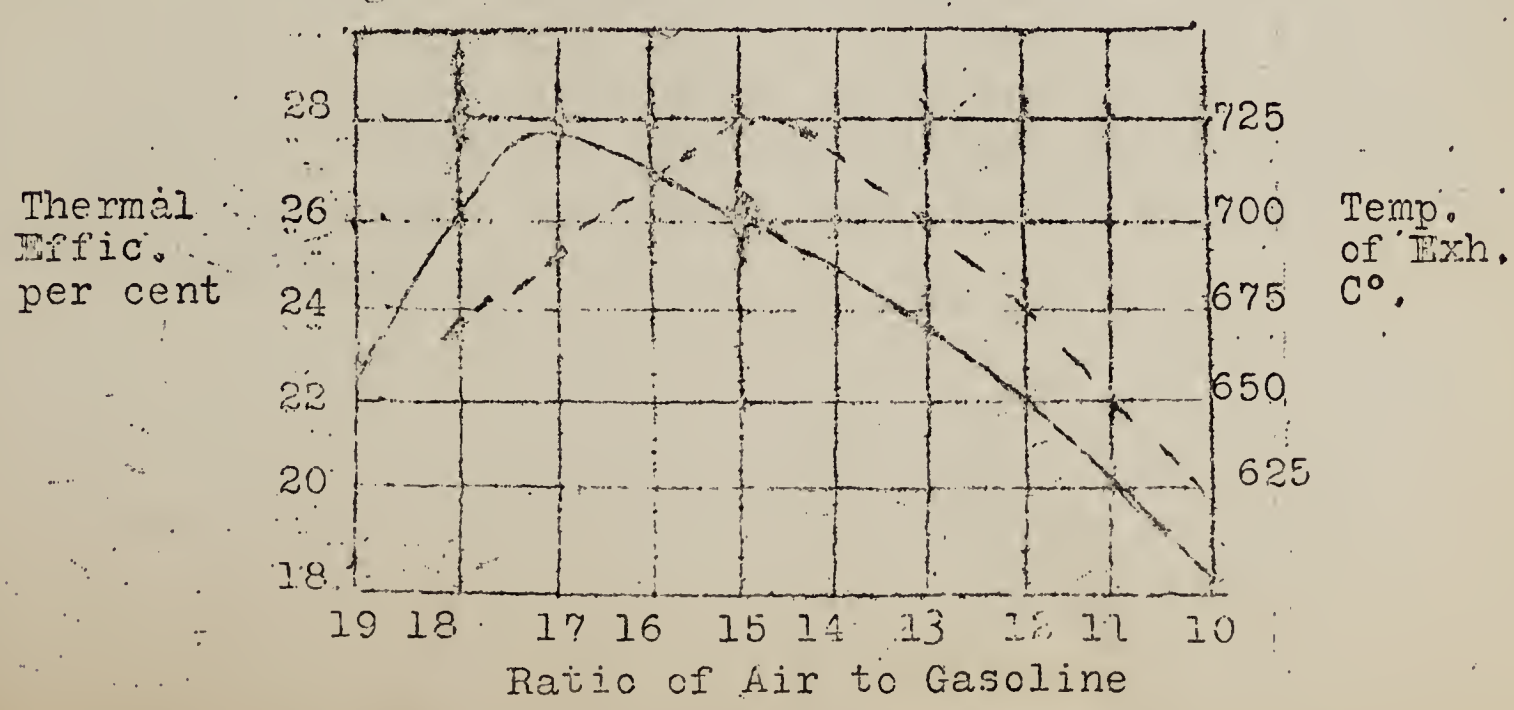
Since the quantity of fuel supplied represents a certain amount of heat energy, the thermal efficiency depends on amount of fuel supplied and on percentage of this fuel that is actually burned successfully.

If the mixture is too rich, there will result incomplete combustion of part of the fuel; part of the fuel will pass into the exhaust without yielding its maximum heat energy,- therefore, lower thermal efficiency.



The curve shows the relation between fuel wasted and carbon monoxide present in exhaust.

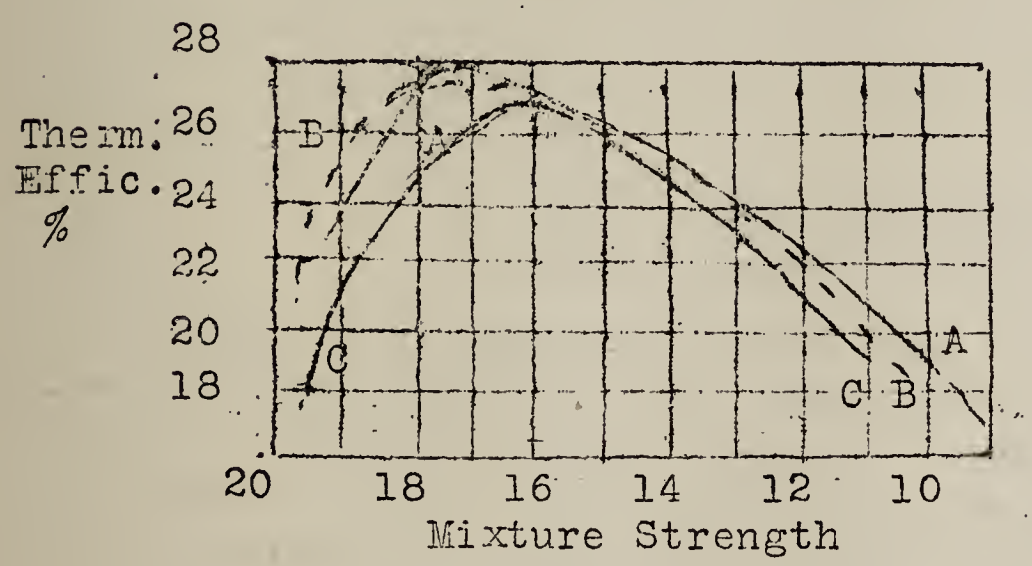
Efficiency with weak mixtures. With over-weak mixtures the governing influences are slowness of combustion and heat absorption by the surplus air present. The curve shows how thermal efficiency changes with different mixture strengths.



The results show that the mixture strength that gives the maximum thermal efficiency is weaker than the proportion that gives complete combustion. Therefore, to obtain highest fuel economy the engine should be run with mixtures somewhat on the weak side of that for complete combustion.

Curve, dotted line, shows measured exhaust temperature for different mixture strengths. Discussion of reason why weaker mixture gives highest thermal efficiency.

Curve showing effect of compression on thermal efficiency and mixture strength.



A, B, and C, have compression ratios of 4.71, 4.35 and 3.92. Same engine, running at same speed.

Curves of same general form as the previous one, although there is a slight tendency with high compressions for the maximum efficiencies to occur with mixtures a little weaker.

This effect of compression must not be confused with the direct influence of amount of compression on efficiency

Nature of Fuel. Practical working conditions must be considered. Compression pressures for highest thermal efficiency; fuel consumption. Relative costs. Since thermal efficiency varies inversely as the product of the quantity of fuel used and its thermal capacity, if the latter is small, the former must be high for same efficiency and horsepower.

Influence of Compression on Efficiency.

From both the theoretical and practical considerations, the thermal efficiency of an engine increases with the compression pressure for the same speed and mixture strength.

From the theoretical aspect, $E = 1 - \left(\frac{1}{\text{comp. ratio}}\right)^{n-1}$

so, if the gas could be compressed indefinitely the efficiency would become 100%.

From the practical point of view, an increase in compression ratio affects the thermal efficiency in several ways.

The losses of heat from the explosive mixture depend, among other things, on:

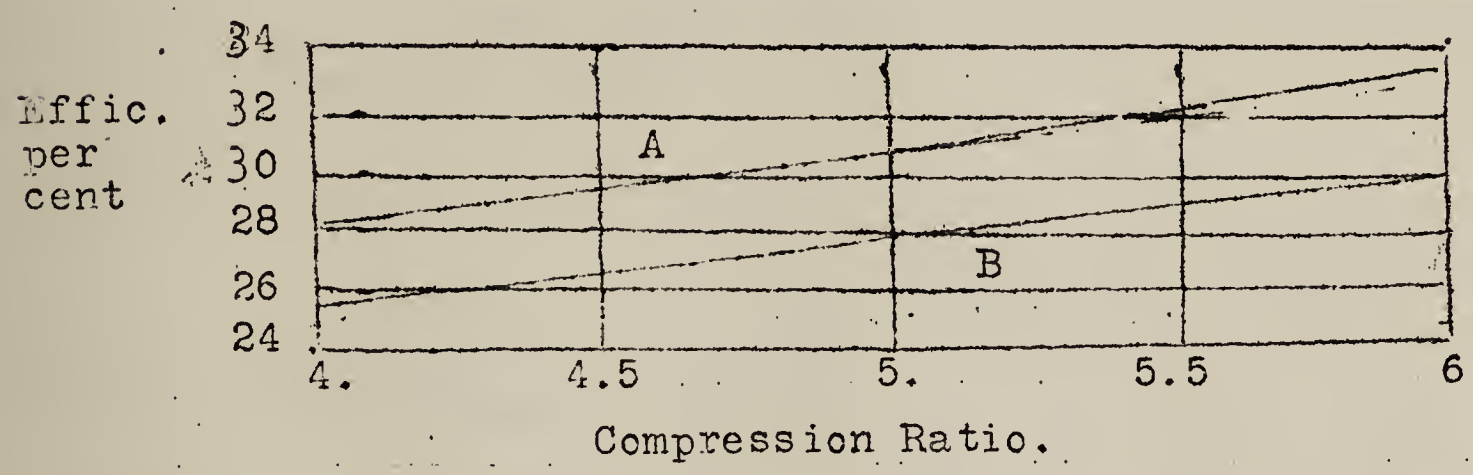
1. The total internal area of the exposed walls,
2. The length of duration of the heat loss,
3. The density of the charge.

There is a decrease in the heat loss from the charge to the cylinder walls as the compression presssure is increasing up to a certain limit; there is a smaller amount of residual exhaust gases, which increases the efficiency; the higher the compression pressure, the quicker the explosion period, and the higher the pressures attained; as the combustion ratio is increased the density of the charge goes-up, and the temperature of combustion increases.

The chief beneficial influence of compression on engine efficiency is to cause the maximum explosion pressure and the mean effective pressure to be higher, due to the greater range of expansion.

The effect of higher compression is to enlarge the indicator diagram for same fuel consumption at same terminal pressure. Tests of Dr. Watson, (page 45) show increase in thermal efficiency with increase of compression.

Ideal Attainable Standard Thermal Efficiency; proposed by Ricardo. Air standard efficiency reduced 34%, so that possible attainable standard is 66% of the air standard.



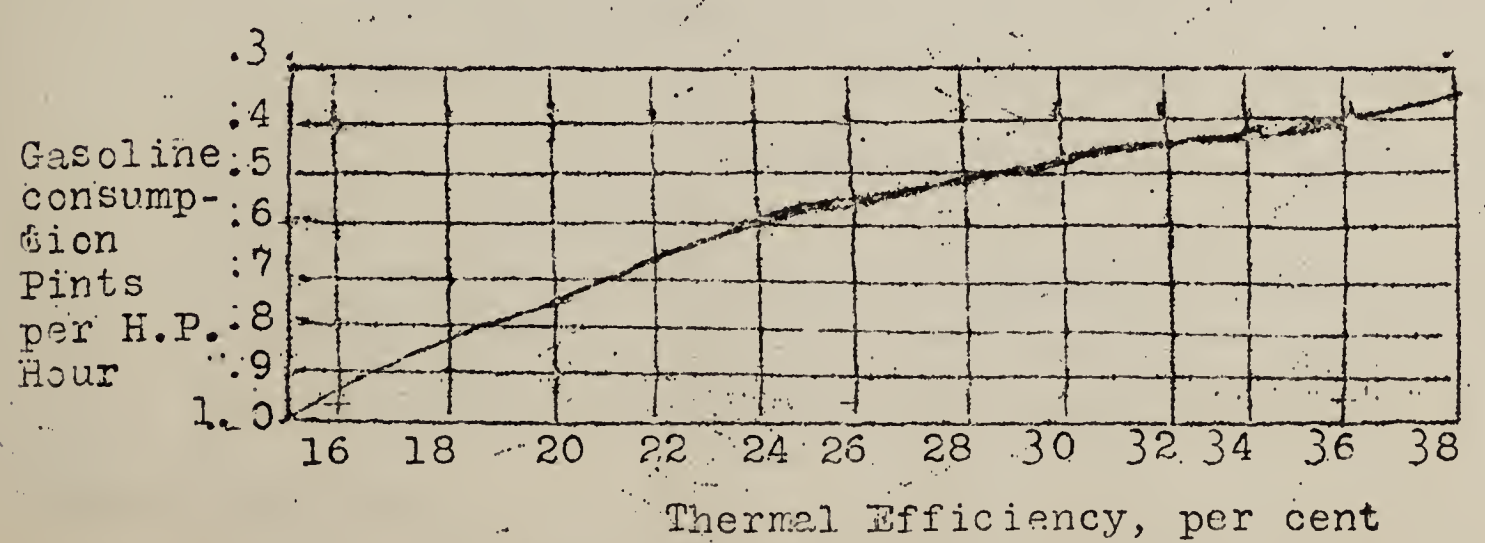
Line A represents proposed ideal attainable standard; line B, the corresponding ideal attainable brake thermal efficiency.

The 34% reduction was made-up of the following allowances:

16% for loss due to change of specific heat
15% for loss of heat to cylinder walls
1% for loss due to explosion not being instantaneous
2% for loss due to early opening of exhaust valve

The curves give results which appear to agree fairly well with actual performance.

Relation between gasoline consumption and corresponding thermal efficiency is shown in the curve which is based on assumption that 1 pound of gasoline = 18,290 B.T.U. = 14.5×10^6 ft. pounds.



Effect of Shape and Size of Cylinder and Combustion Chamber.

Effect of Shape. As attainable explosion pressure depends somewhat on amount of heat abstracted by walls, the shape of combustion chamber influences the thermal efficiency. If area is large compared to volume, heat loss to walls will be greater and pressure attained will be lower.

Table given ratio of combustion chamber area to clearance volume.

		Volume of Clearance	Ratio of Surface to Volume
Actual T-head Engine	A	184 c.c.	2.01
	B	203 "	1.86
	C	233 "	1.69
Equivalent Hemispherical Shape	A	184 "	0.99
	B	203 "	0.94
	C	233 "	0.88

Ratio of surface to volume of hemispherical shape is about 50% of that for T-head.

Several successful engines have combustion chambers which approximate the hemispherical shape.

Nature of cylinder walls. Loss of heat to walls depends on the condition of their surfaces. Heat radiated from polished surfaces less readily absorbed by walls,- higher thermal efficiency. With rough, unfinished walls absorption of heat is greater. Rate of heat loss to carbon-coated surface is reduced and less heat is abstracted from exploding charge.

Effect of Size. Heat loss varies as ratio

$$\frac{\text{surface}}{\text{volume}} = \frac{1}{\text{diameter}},$$ inversely as their diameters.

Efficiency becomes higher as diameter is increased. Efficiency of engines of small sizes would be correspondingly low; relative fuel consumption would be higher.

Stroke-Bore Ratio.

Relative Thermal efficiency of long and short stroke eng

For such stroke-bore ratios as are practically usable in airplane engines, the difference in thermal efficiency is practically negligible.

Off-Set Cylinders.

Effect of Speed on Thermal Efficiency.

As speed of a gasoline engine is increased up to a certain limit, the thermal efficiency increases. At high speeds there is less time for heat exchange between the charge and cylinder walls,- better efficiency. Variation of efficiency with speed is nearly the same for different compression ratios and different mixture strengths over a wide range. Thermal efficiency of small engines due chiefly to the very much higher speed at which they can run.

Effect of Throttling on Efficiency.

Thermal efficiency increases as speed is increased; and highest thermal efficiency is obtained when engine is throttled to about three-fourths power.

Practical considerations that fix upper limit of speed.

Best thermal efficiency is attained before highest speed is reached; at this speed fuel consumption is lower. Often desirable from considerations of engine rating, power for weight, to obtain maximum power for a given cylinder size.

Effect of Valve Timing on Efficiency.

Time of valves has noticeable effect on thermal efficiency. Amount of fresh charge depends on period and amount of inlet-valve opening; time of opening exhaust valve governs ending of expansion and amount of heat lost to the exhaust. Time of closing exhaust valve influences amount and temperature of exhaust gases which affects suction temperature. Particular attention given to effect of time of closing inlet-valve.

Actual Thermal Efficiency

Greatest possible thermal efficiency of gasoline engine working under ideal conditions (no losses to water jacket, etc.) is 20% to 25% less than the air standard efficiency. So; with engine working with compression ratios between 5 and 6, the highest efficiency that could be expected to obtain would be about 38%.

Volumetric Efficiency

The quantity of charge is always less than the theoretical quantity. The ratio of the actual quantity to the theoretical quantity is called the "Volumetric Efficiency".

The quantity of fresh charge induced into the cylinder depends on several factors,-

1. Frictional resistance offered to the flow of gas
2. Velocity of the gas
3. Heating of the charge
4. Mechanical suction of piston
5. Timing of the valves.

The frictional resistance encountered by the charge is dependent on the following factors:

1. The design of the carburetor.
2. The length and dimensions of the inlet pipe.
3. The number, size and placement of the valves.
4. The number of bends in the induction line.

Ordinary carburetors make use of venturi or choke tubes through which the charge is given an increase of velocity of at least 100%. Throttling, or wire-drawing, of the air supply occurs also at other parts of the carburetor.

Short pipes of ample size are necessary for high volumetric efficiency.

Valves should be so designed that there is no appreciable increase of velocity past them, and also no undue heating of the charge.

Each bend in the induction line is a source of loss, due to eddies and turbulence.

Engine does not fill entirely its stroke column with a fresh charge.

Density of Charge

The density of the charge in the cylinder is a direct measure of the volumetric efficiency.

Example. If the temperature of the inlet charge is 135° F, then its density would be

$$\frac{32 + 459}{135 + 459} = \frac{491}{594} = 0.827 = 82.7\%$$

With actual engines, however, it is not possible to obtain atmospheric pressure at the end of the suction stroke, so the actual volumetric efficiency will always be less than this ideal value.

Ricardo has proposed a reduction of the ideal volumetric efficiency by 5% as a standard of comparison with actual engines,- reduction is equivalent to reducing the terminal pressure to 14.3 pounds.

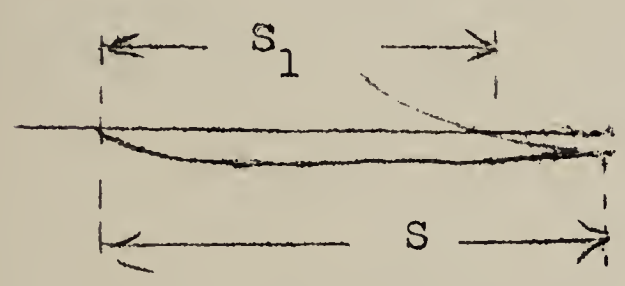
With this as a standard, the ideal volumetric efficiency in the example would be 0.95 X 0.827, or 78.6%.

This value corresponds to a gas velocity of about 120 feet per second in a short pipe.

Volumetric efficiency is dependent on the density of the fresh charge; that is, on the temperature and pressure of the charge in the cylinder.

The maximum possible amount of useful work depends on the quantity of the charge in the cylinder at the end of the suction stroke; and the quantity of this charge will be initially governed by the amount of oxygen present in the induced charge.

The volumetric efficiency can be obtained from the indicator diagram if the point of crossing of the compression line with the atmospheric line can be determined.



$$\text{volumetric efficiency} = \frac{S_1}{S}$$

Velocity of Flow.

The mean velocity of flow for small differences of pressure, can be found approximately from the formula

$$v = \sqrt{\frac{2g}{d}\ (P - P_s)}$$

where d is the density of the gas (pounds per cu.ft.) and P the atmospheric pressure and P_s the suction pressure in the cylinder. If the pressures are in pounds per sq.ft., the velocity will be given in feet per sec.

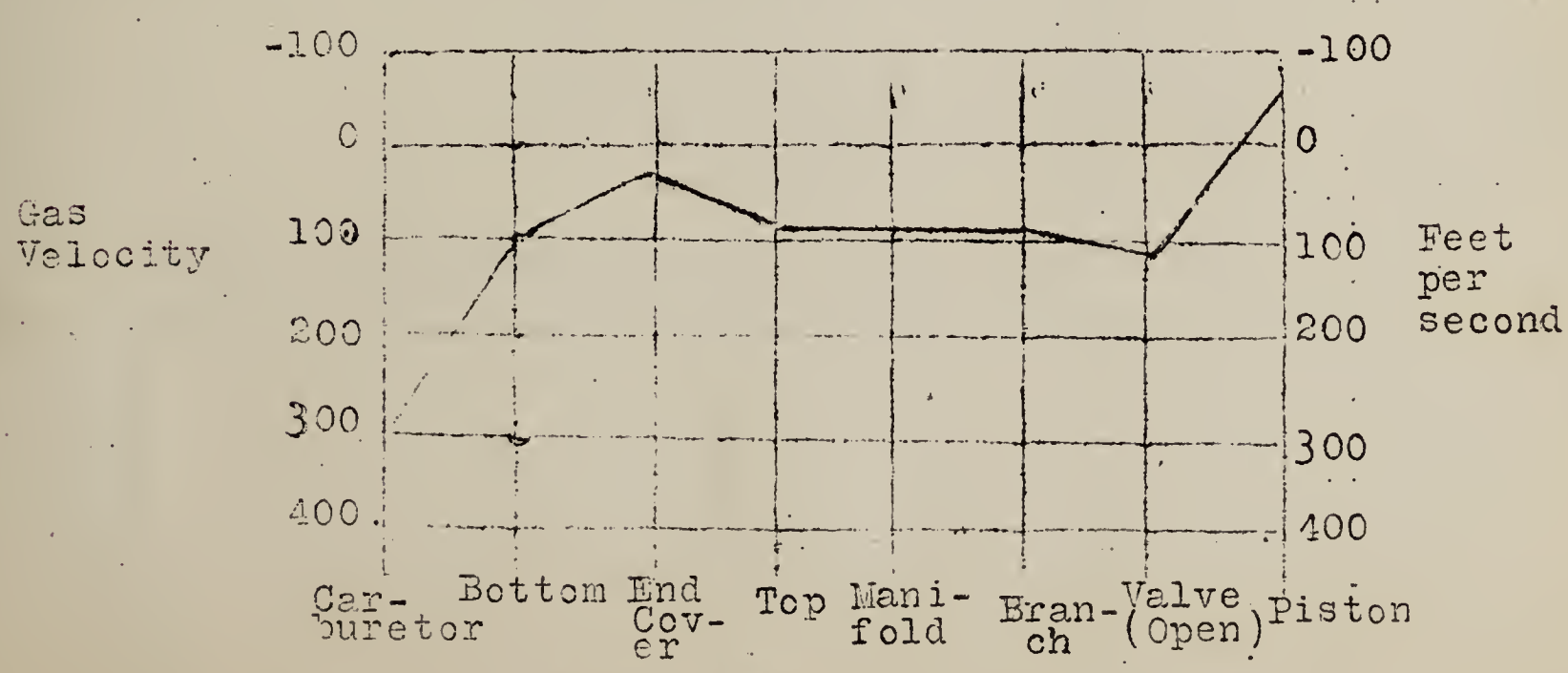
Example. Taking the difference of pressure between the atmosphere and that in the cylinder during the suction stroke as one pound, $P_s = 13.7$; and assuming the density of the charge to be 0.09 pounds per cu.ft.

$$v = \sqrt{\frac{64.32}{0.09}(14.7 - 13.7) \times 144} = 320 \text{ ft. per sec.}$$

This theoretical velocity is higher than would be attained in an actual engine, as no allowance has been taken of the effect of resistance, eddies, obstructions, and skin friction.

Velocity of induced charge depends on the difference of pressure between the atmosphere and that in the cylinder during the suction stroke; but, for high volumetric efficiency, it is imperative to keep this velocity as low as possible, therefore, the pressure difference must be small. The resistance due to eddying, bends, and skin friction vary approximately as the square of the velocity. Tests have shown that at high speeds, the resistance may vary at a higher ratio of velocity than the square.

In order to keep the inlet velocity as small as possible, large valves, or a multiplicity of valves, combined with fairly straight and smooth pipes and passages are used. The inlet velocity varies from 125 to 150 feet per second according to the engine speed. A suitable velocity for very high engine-speeds, appears to be between 250 and 275 ft. per second.

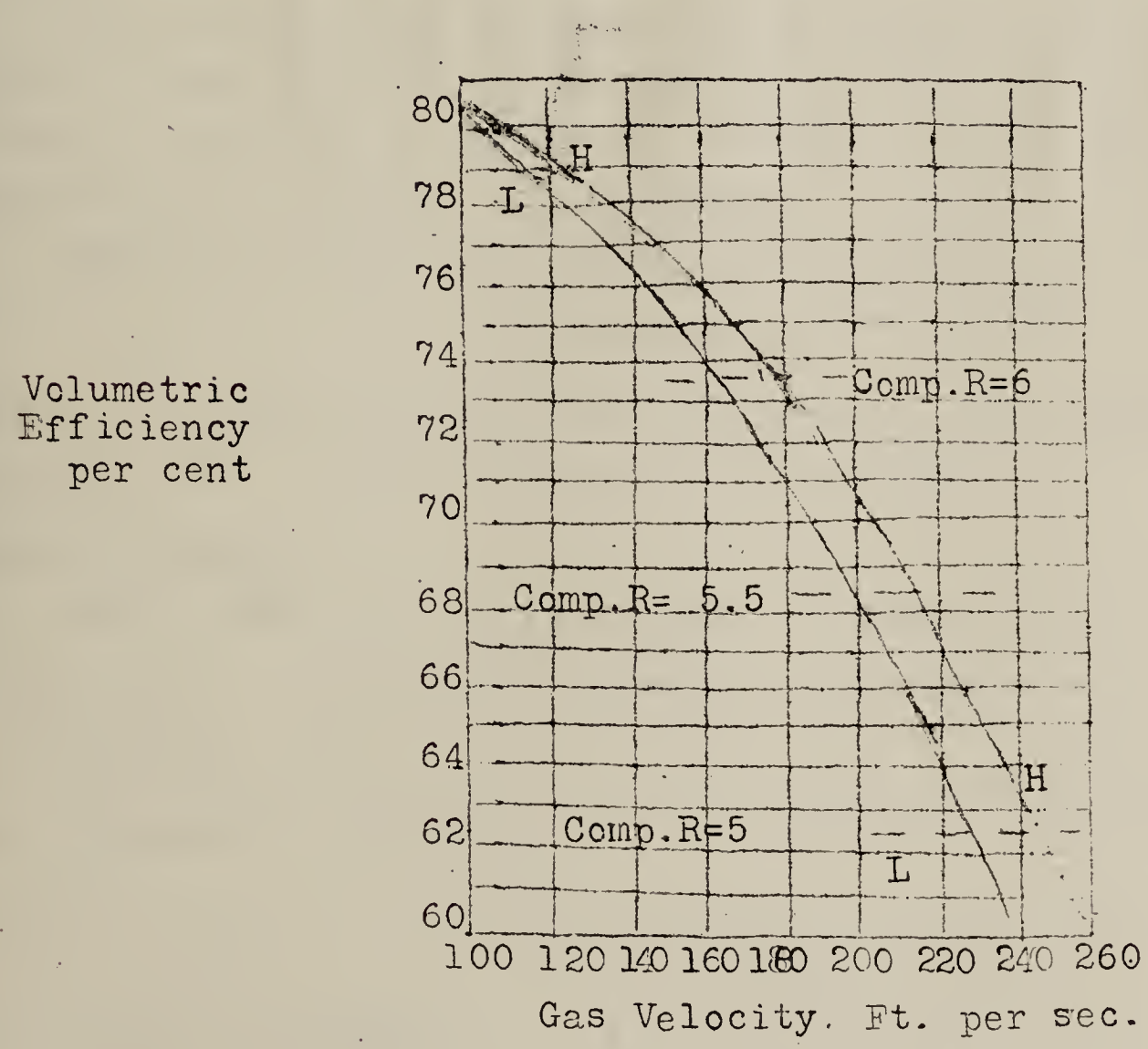


Length of inlet pipe = 45 in.

Diagram illustrating gas velocities in different parts of induction line for a R.A.F. 1b. aeroplane engine. 8-cylinder V-type air-cooled, bore 4-1/8", stroke 5-1/2", compression ratio 4.5. Power out-put, 105 b.H.P. at 1600 r.p.m.

Ricardo's curves give a hypothetical relationship between the volumetric efficiency and the gas velocity for short inlet pipes. Allotment of a volumetric efficiency of 80.5% for a gas velocity of 100 ft. per sec. Loss of 19.5% was placed as follows:

Least loss to heating at inlet valve 17.3%
" " " pipe resistance 2.2%



Curve H is for engines with valves in the head.

Curve L is for engines with valves in side pockets.

Skin Friction

Frictional resistance to gas flow may be represented by

$$f = \frac{c\,l}{m}\,v^2$$

where c is coefficient dependent on nature of surface; l, length of pipe; m, ratio of area of section to perimeter; v, velocity of flow. For cirular pipe

$$f = \frac{4\,c\,l}{D}\,v^2$$

where D is diameter of pipe.

The formulae are approximate for velocities less than 150 ft. per sec.

Resistance varies directly as the length of the pipe and the square of the velocity, and inversely as the diameter or the ratio of area of section to perimeter. Effect of loss in gas flow is to cause a loss in weight of the charge.

Formula of National Physical Laboratory

$$F = 0.0000082\,A^{0.93}\,V^{1.86}$$

where F is the friction force in pounds, A the area of surface in sq.ft. and V the velocity in ft. per sec.

Effect of Bends.

Effect of bends is equivalent to increased frictional resistance. The sharper the bend, the greater the equivalent frictional loss.

The manner in which the loss varies with the radius of the bend is shown in the table.

Radius of Bend	11	7	5	2	1	.75	.5
Equivalent Resistance expressed in length of straight pipe	4.1	6.1	7.7	9.0	17.3	34.1	118

Other Causes of Loss in Induction Line.

Loss due to sudden changes of section.
Loss due to obstructions, such as carburetor nozzle, valve stem, ledges, etc.
Loss due to heating of charge by hot walls and piston and to work done on gas by piston.

So far as frictional losses are concerned, back-pressure effect of exhaust gas flow follows same conditions as the induction flow.

Gas Flow Through Valves.

Design of valves has important bearing on volumetric efficiency; Effect of friction at high velocity of flow is serious.

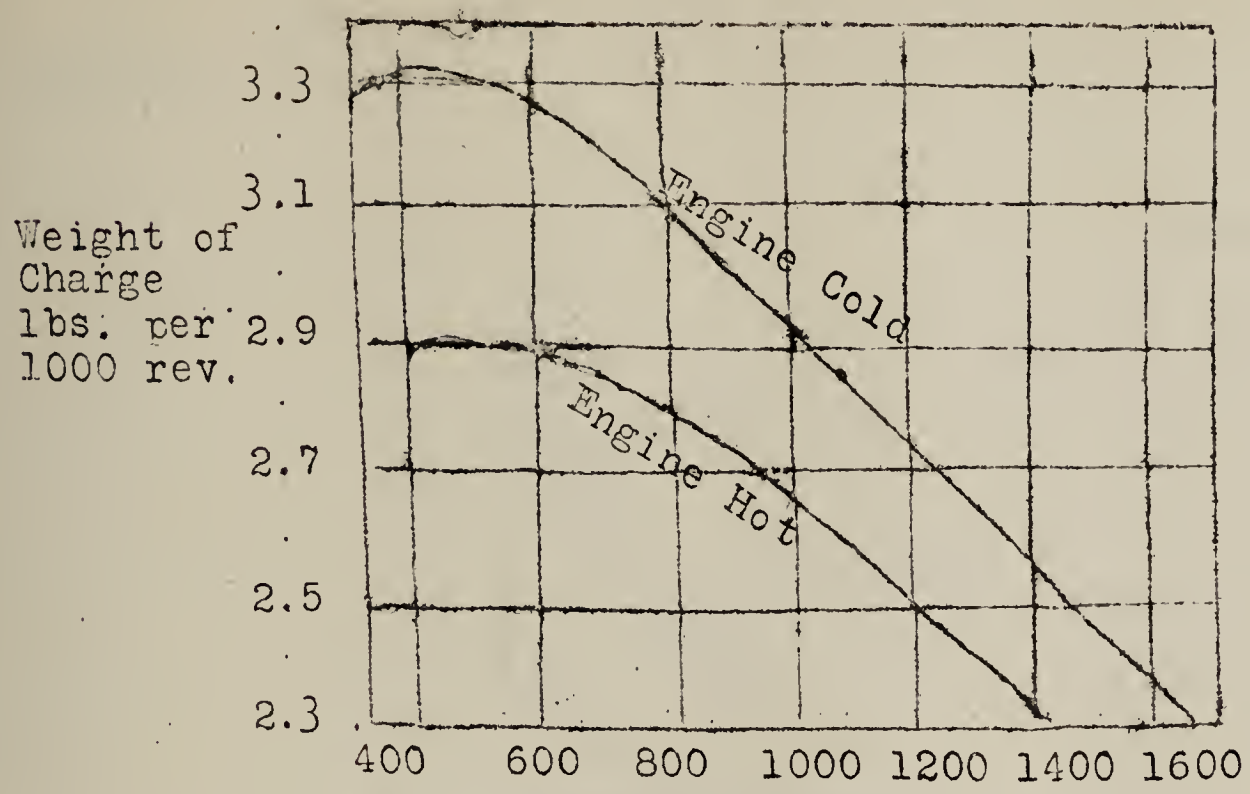
Placement of Valves. Best arrangement for volumetric efficiency is overhead valve type,- charge comes straight into cylinder with practically no change of direction. With side-by-side valve type, charge has to make two bends of about 90° to reach the same place in the cylinder, - loss due to bends at high gas velocity is considerable. Another reason; with side-by-side valves the mixture has to pass over a larger area of combustion chamber before it arrives in the cylinder, and the charge becomes heated more, and loses in density.

Heating of the Charge

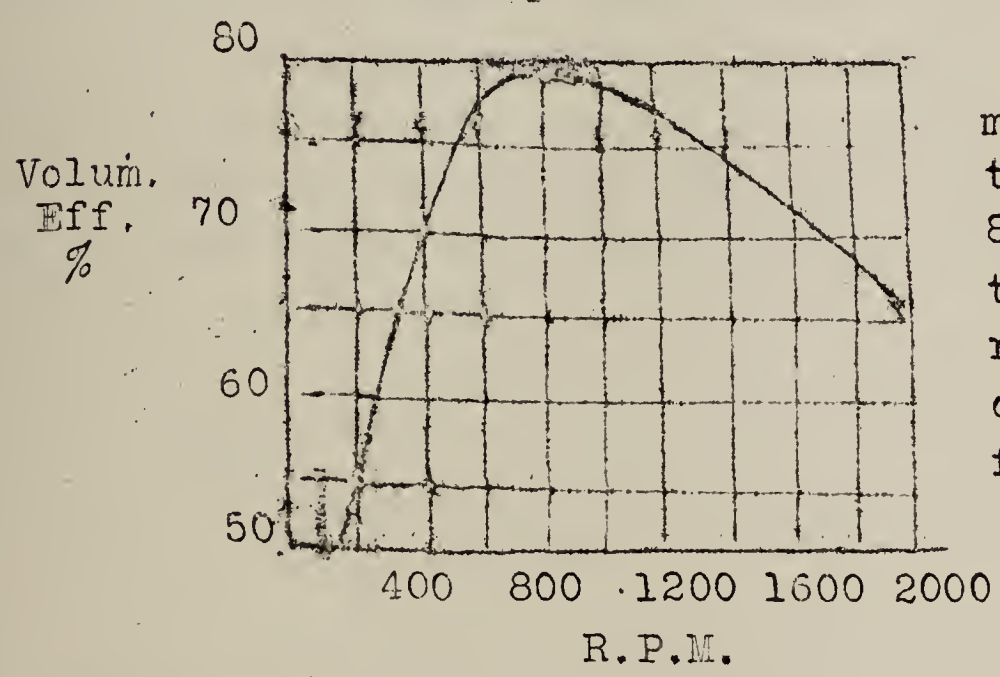
Entering charge becomes rapidly heated. Effect of premature heating of charge is to cause gases to expand. Also charge becomes heated by mixing and contact with exhaust products of previous stroke.

Effect of hot engine in reducing the charge shown by curves of tests on four-cylinder automobile engine. The curves give an increase in quantity of charge with cold engine of 10% to 12% for speeds above 800 r.p.m. Charge falls-off rapidly with increase of speed.- volumetric efficiency decreases as engine speed increases.

Effect of Temp. on Volumetric Effic.



Curve showing volumetric efficiency as a Moline-Knight engine at different speeds.



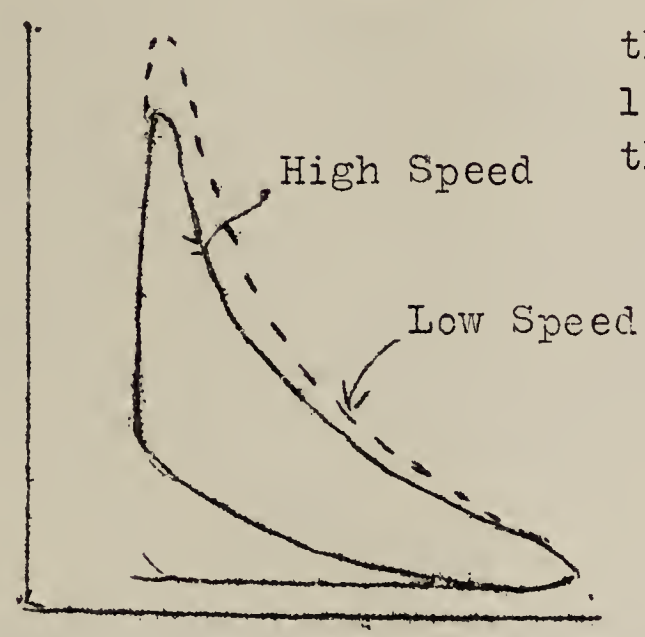
Generally speaking, in modern motor-vehicle practic the charge efficiency is fro 80% to 85% at 1000 r.p.m., a the efficiency falls-off regularly with increased spee on account of the increased frictional resistance.

Effect of Quantity of Charge

Effect of running an engine practically throttled is somewhat the same as an engine showing poor volumetric efficiency. Maximum power is determined by the amount of charge which can be drawn into the cylinder.

Effect of low volumetric efficiency at high eng speeds is to lower the compression pressure;- the effectiv pressure on the piston is therefore less.

Compression effect is represented in the indicator diagram.



Difference in area of the two diagrams represent loss in effective pressure that accompanies high speed.

When an engine gives low volumetric efficiency, any efforts that are made to raise the volumetric efficiency that result in still higher speeds, and higher explosion pressures, may not be beneficial to the working parts of the engine; which may have a strength to resist only the lower pressures and temperatures.

It is common practice in the design of high-speed engines to fix the valve area and valve timing, placement of valves, etc., so that the volumetric efficiency is high at the working speeds; and it does not always follow that the engine is flexible at all speeds.

Temperatures within the Cylinder

Temperatures existing within the cylinder of a gasoline engine can only be estimated approximately.

Direct measurement of temperatures during the working cycle is a difficult problem,- because of high explosion temperatures and rapid temperature changes.

In making study of the behavior of the gases it is essential to know with some degree of accuracy the temperature, at least, at one important point in the cycle from which the temperatures and pressures at other points may be calculated.

Suction Temperature. If the exhaust temperature is known or can be determined, it is possible to calculate the suction temperature with a fair degree of accuracy from the relation.

$$T_s = \frac{ntT_e}{(n-1)T_e + t}$$

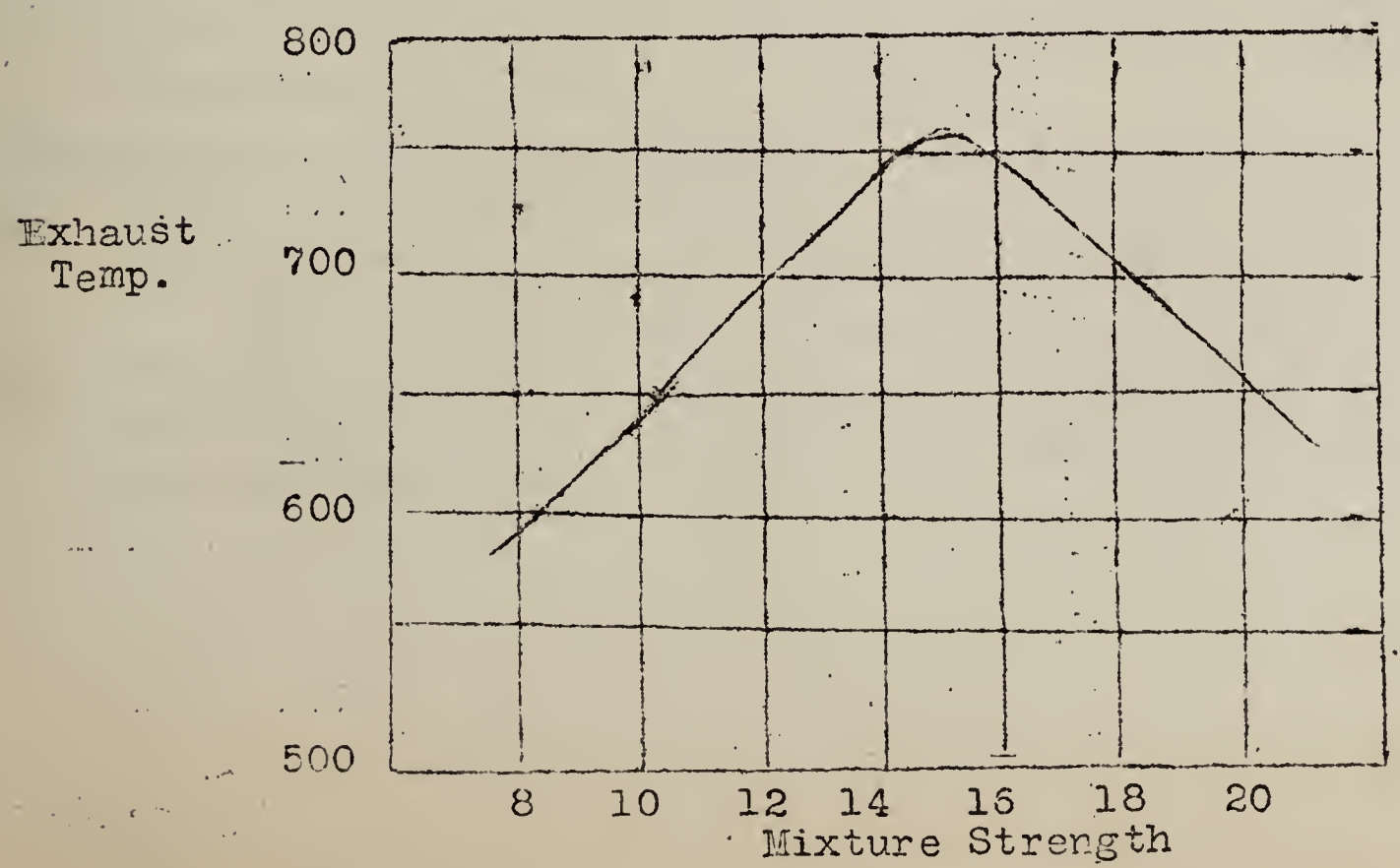
Where t is the temperature of the atmosphere and T_s and T_e the temperature of suction and exhaust respectively.

The following exhaust temperatures have been suggested as being fairly representive. For compression ratios 4, 5 and 6, the temperatures are 1530°, 1470° and 1440° F.

Example. If temperature of entering air is 60° F and compression ratio 5,

$$T_s = \frac{5(60 + 459)(1470 + 459)}{4 \times 1929 + 519} = 608° \text{ abs.}$$

The mean temperature of the exhaust gases for different mixture strengths obtained from an automobile engine running at 1100 r.p.m. is shown in the curve.



The exhaust temperature also depends on the compression ratio, engine speed, and general design of the engine, so it may be expected to vary with different engines.

Measurements have shown that there is very little difference in temperature between the top and bottom of the cylinder barrel. This difference may be 35° F. and it becomes smaller as the speed increases; the value may be as high as 115° F. with air-cooled engines.

The mean temperature of the exhaust valve is probably about 700°F; the mean temperature of the inlet valve probably lies between 250° F. and 450° F. The mean piston temperature may be about 600° F. These figures must be taken as provisional only.

With airplane engines there is likely to be a difference in the temperature between the front and the back of the cylinder; this difference of the maximum temperature may be as much as 100° F.

Tests of Dr. Gibson and others have shown that the maximum temperature of the combustion head should not exceed 550° F. for air-cooled engines. Temperatures above 570° F. are liable to distort the cylinders and crack the valve seats; pre-ignition trouble also occurs. The best results in gasoline consumption and power output with air-cooled engines appear to be obtained with maximum combustion-head temperatures of from 400° to 450° F. If the engine is larger than 4-in. bore, higher values are allowable.

The exhaust valve temperature should not exceed 1300° F. It is possible to use a working temperature of 1200° F. quite satisfactorily with 1.5 in. valves.

<u>Temperature Variation with Speed</u>. The temperature of the cylinder at any part increases with the engine speed,- variation is more marked at the lower speeds than at the higher speeds.

Table giving results of test made at different speeds with full load under a cooling wind of 53.5 miles per hour at 22° C. Engine had aluminum cylinder, 4.5" bore, 5.5" stroke, and compression ratio 4.7.

Temperature on Side of Combustion Space

Engine Speed in R.P.M.	B.H.P.	Temperature °C.
800	10.2	100
1,000	12.8	103
1,200	15.4	124
1,400	18.0	123
1,600	19.7	136
1,800	20.6	138

Effect of Compression Ratio

For every engine there appears to be a certain compression ratio which gives a minimum wall temperature, and that for higher or lower compression ratios the temperatures increase.

Table showing the effect of compression on temperature of cylinder barrel as obtained from tests on an aluminum air-cooled cylinder of 4" bore and 5.5" stroke under maximum load in a cooling wind of 64 miles per hour. Gasoline consumption is also given.

Compression Ratio	Brake M.E.P.	Gasoline Consumption per B.H.P. per hr. Pounds	Mean Temperature of Cylinder Barrel in ° C.	
			Top	Bottom
4.6	116.2	0.530	180	105
5.0	119.3	0.507	170	95
5.4	122.0	0.490	157	89
5.8	125.0	0.475	154	85
6.2	129.0	0.480	183	110
6.4	123.0	0.520	212	135

The results show that the minimum wall temperatures and lowest gasoline consumption occur for the compression ratio 5.8. The practical limiting value seemed to be about 6. For compression ratios above 6, there was excessive charge heating and cylinder temperatures; occasional "pinking" occured with a compression ratio of 6.2; and there was heavy knocking with a compression ratio of 6.4.

Increase in mean effective pressure and thermal efficiency with increase of compression ratio is also shown.

Effect of Mixture Strength

Greatest amount of heat is liberated from mixtures of about the correct proportion of air to gasoline, - cylinder temperatures would be a maximum under such mixture strength.

The results of measurements of exhaust gases; given on page 58, support this view. In the case of weaker mixtures the hot gases give a somewhat higher exhaust valve temperature, tending to cause overheating.

Table giving Exhaust-Valve Temperatures for different mixture strengths obtained from the air-cooled aluminum cylinder engine, 4." bore, 5.5" stroke, compression ratio 5.5.

Air/Gasoline Ratio	Brake M.E.P.	Gasoline Consumption in lbs. per B.H.P. hour.	Exhaust Valve Temp. at 1400 r.p.m. ° C.
11.1	122	0.622	706
11.9	122	0.589	717
13.8	119	0.515	747
15.2	116	0.480	752
15.7	114	0.470	747

The maximum cylinder-head temperature was 237°C with a mixture of about 13.0, falling to 200° C for a ratio of 10.5 and to 215° C for a ratio of 15.4. The air temperature was 23°C.

The mixture proportion that gave the greatest power output was about 13.5 which is about the ratio for the maximum cylinder-head temperature, and the same as that giving the highest cylinder-liner temperature.

Effect of Timing of Ignition

Point of ignition has marked effect on the character of combustion; very late ignition prolongs combustion, very early ignition causes heat accumulation, - the effect of either tends to increase cylinder temperature.

Table giving results of tests by Dr. Gibson on an aluminum-headed air-cooled cylinder, 5.5" bore by 6.5" stroke.

Engine was run at 1450 r.p.m. in a cooling wind of 80 miles per hour. The only variable was the spark advance.

Spark Advance Degrees	B.H.P.	Gasoline Consumption in lbs. per B.H.P.-hr.	Temperature of Center of Combustion Head °C
21	30.5	0.515	215
26	31.2	0.498	215
30	31.6	0.492	221
36	31.6	0.492	245
40	31.0	0.500	275

When the spark was advanced far enough to cause "pinking", the temperature rose appreciably.

Table giving results of tests by Dr. Gibson on an Hispano-Suiza water-cooled engine with aluminum cylinders and steel liners using two spark plugs to a cylinder.

	Cylinder Temperatures			
	Cylinder		Another Cylinder	
Ignition	1" below upper surface of piston	3.3" below upper surface of piston	1" below upper surface of piston	3.3" below upper surface of piston
Two Spark Plugs	241°C	191°C	182°C	138°C
One Spark Plug	172°C	137°C	144°C	114°C

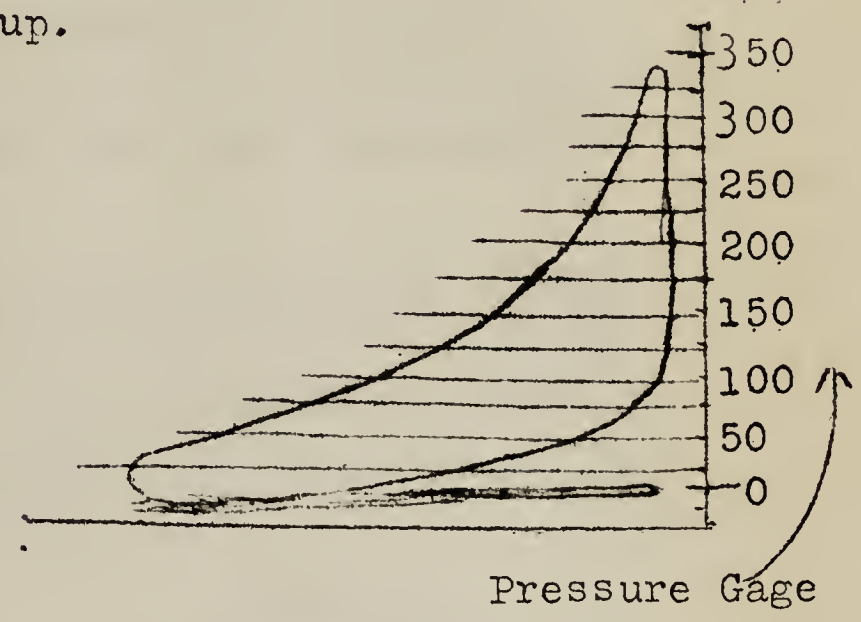
Piston temperature was 85° to 100°C higher than that of the cylinder barrel, and for wall temperatures above 180° C. piston burning occurred. With cast-iron pistons the temperature may be from 120° to 180° C. higher than when aluminum pistons are used. In a certain engine test the maximum temperature of the aluminum piston was 240° C. while with a cast-iron piston it was 400° C; the cylinder-liner temperature being 145° C.

Pressures in the Cylinder

Frequently necessary to assume or predict the indicator diagram which a new design of an engine would give, in order to compute the piston forces, the forces on the pins and crankshaft, the turning moments, etc.

Several methods for working-up.

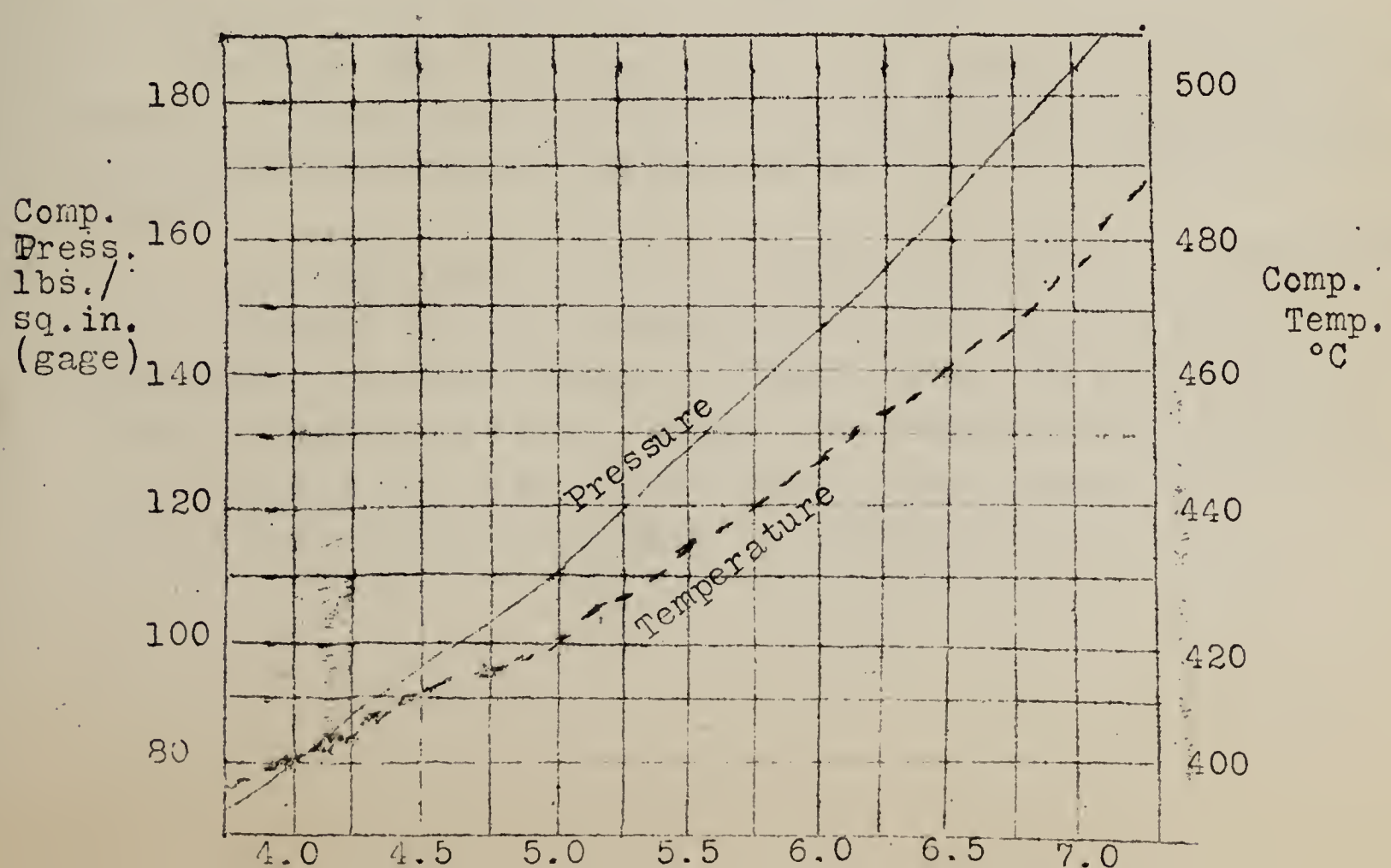
A diagram typical of good performance for a four-cylinder automobile engine having a compression ratio of 4.3.



Correct value of "n" difficult to obtain. Values suggested as appropriate.

For compression curve, n = 1.35
For expansion curve, n = 1.3

Curves (Ricardo) giving compression pressures and temperatures for different compression ratios estimated for the value of n = 1.35. Pressure at end of the suction stroke was taken as 14.45. Results are probably accurate within 2% of a normal engine speed of 1800 r.p.m.



Empirical Relations of Pressure.

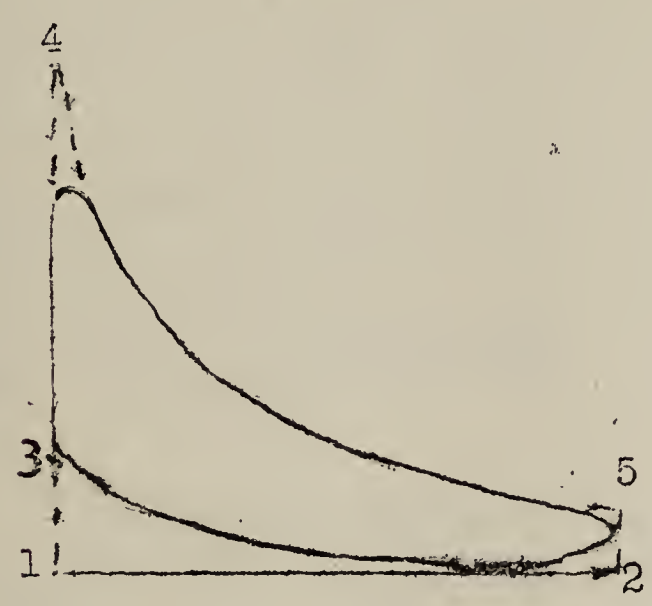
Approximate emperical relationship. Maximum explosion pressure is 3.5 to 4 times the compression pressure.

Mean effective pressure is 0.8 to 1.2 times the compression pressure. Relationships are very approximate.

Assumption Indicator Diagrams.

Method of building-up a predicted indicator diagram from the mean effective pressure. The M.E.P. is either assumed or is taken from test data on similar engines.

The actual indicator diagram of the engine must then give a m.e.p. equal to this value.



Procedure

Area of the indicator diagram =

$$A = \text{area } 1452 - \text{area } 132.$$

$$= \frac{P_4V_4 - P_5V_5}{n - 1} - \frac{P_3V_3 - P_2V_2}{n - 1}$$

and the

$$M.E.P. = k \frac{A}{V_2 - V_1}$$

where k is a constant which allows for the rounding-off of the corners; its value may be about 0.95.

If P_2 and P_3 are known, it is apparent that the solution of the problem involves assuming a value for either P_5 or P_4, which assumption brings us back to the original conditions.

A process which seems as direct and quite as instructive is to proceed with an assumed value for P_4 and by use of the planimeter find the m.e.p. of the diagram that has been drawn. Then by trial and error, adjust the conditions to meet the required m.e.p. Also, under this method, account would be tak of the losses due to suction and back-pressure.

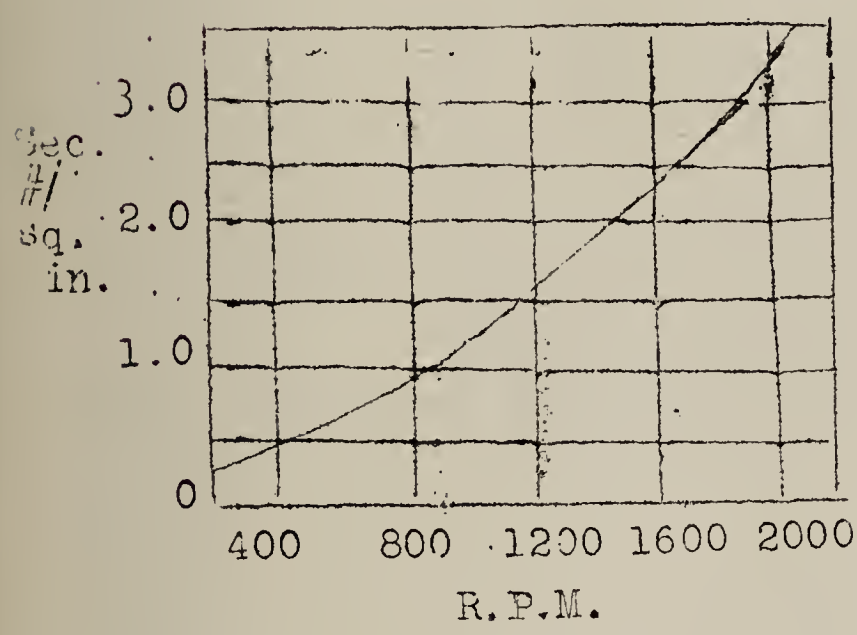
Manifold Pressures.

Pressures in Inlet Manifold. Mean pressure during any fixed condition of running; and pressures existing at different periods during a complete cycle.

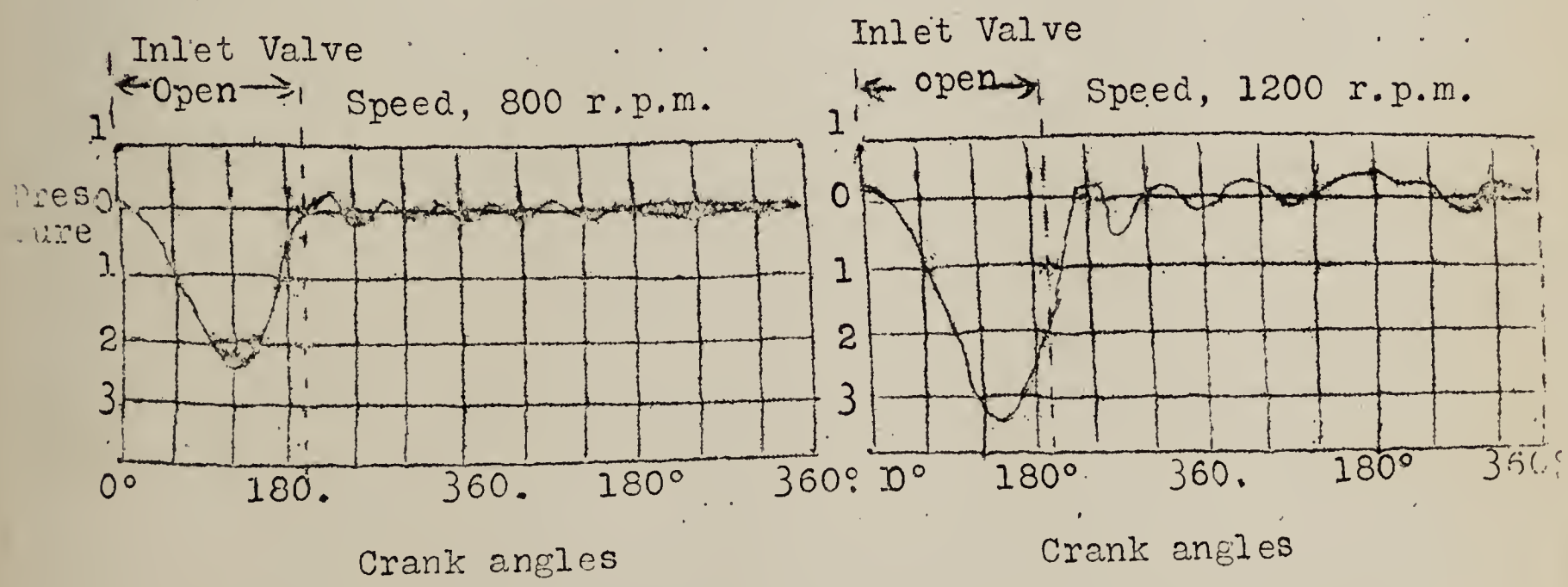
Average pressure is always negative. In ordinary automobile engine 4 pounds may exist at high speeds; low volumetric efficiency accompanies such high suction.

Maximum negative pressure is dependent on design of carburetor, inlet pipes and ports, and is a function of engine speed and valve timing.

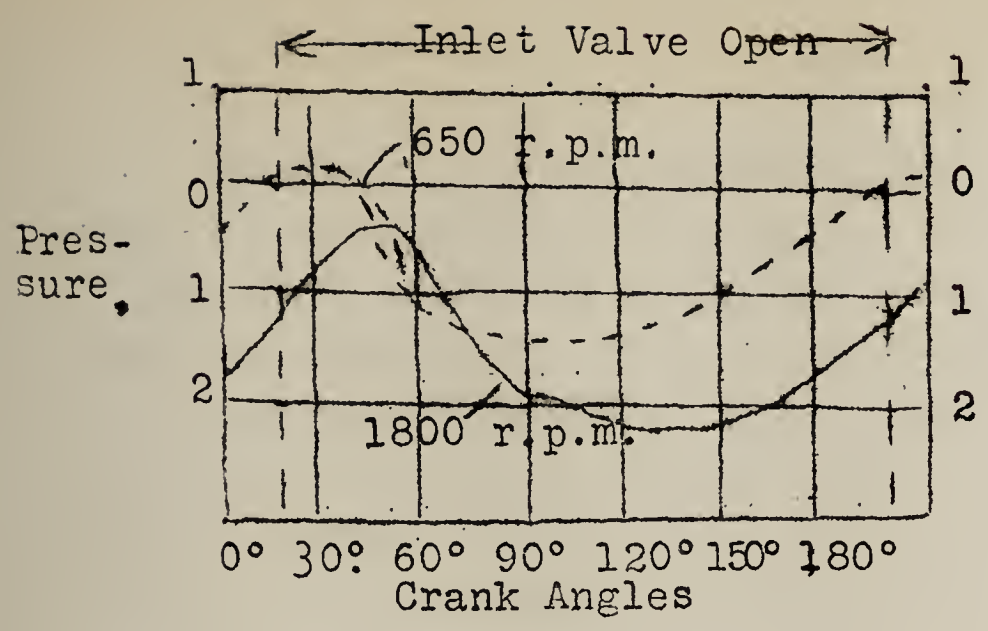
Variation of suction with speed is shown in curve of mean negative pressures at different speeds in four-cylinder automobile engine.



Indicator diagrams from induction pipe of single-cylinder sleeve-valve engine (Dr. Watson).



Effect of increased speed is to increase amplitude of pressure variations.

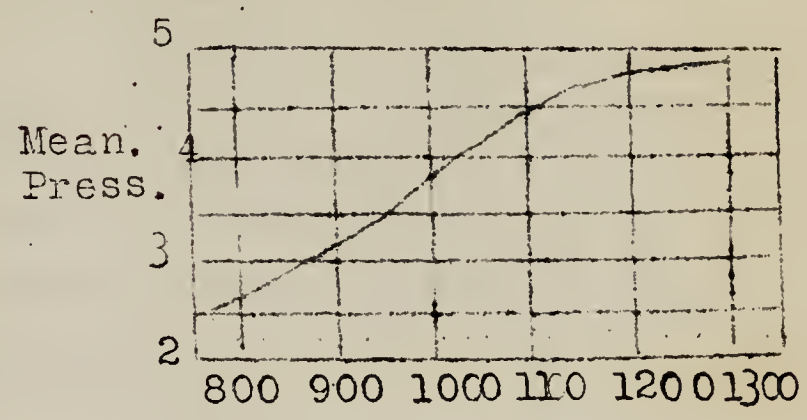


Curves showing variations of pressure with four-cylinder engine. Effect of multiplication of cylinders is to smooth-out pressure variations; more uniform suctions generally occur in these cases. Amplitude of pressure variations also reduced. Greater frequency of cycle changes,- the greater the frequency the smaller the amount.

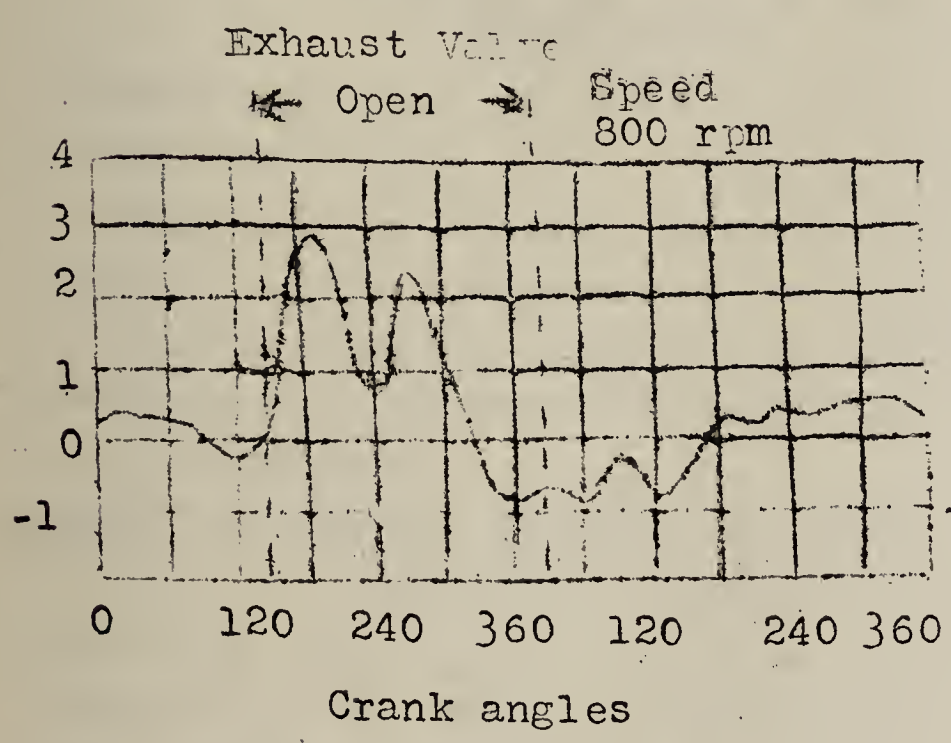
With four-cylinder engine, note rise of pressure in inlet pipe just after inlet valve opens, - due chiefly to inertia of gases. Pressure rise becomes greater the longer the inlet pipe; with very short inlet pipes, no rise of pressure above atmospheric occurs.

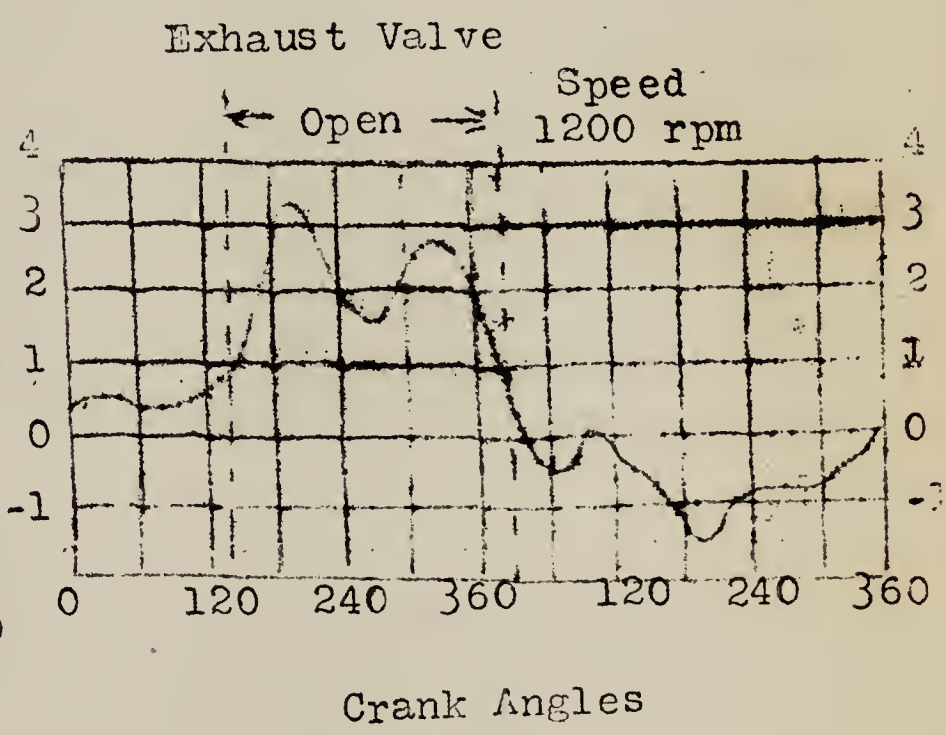
Volumetric efficiency falls-off a little more rapidly than does mean pressure in inlet pipe.

Pressures in Exhaust Manifold. A typical curve showing mean exhaust-pressure variation with engine speed in a single-cylinder engine.



Curves showing the variations of exhaust pressures in the case of the sleeve-valve engine considered in connection with the inlet pressures.

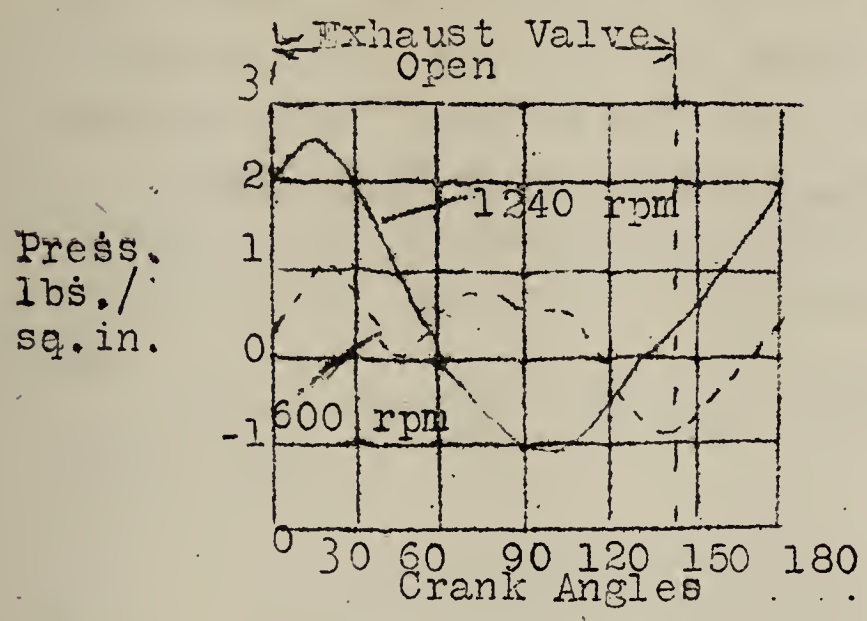




Amplitude of pressure variations is greatest during period of exhaust-valve opening, and just before the valve closes there is a fairly rapid drop of pressure below atmospheric. Negative pressure effect due to inertia of the rapidly moving exhaust gases.

Scavenging action. Length of exhaust pipe plays an important part in scavenging effect. There is a certain length of pipe for a given engine which will give pressure waves in resonance with the engine speed.

Curves showing variation of pressure in exhaust pipe of four-cylinder engine.



Period of surgings smaller than with the single-cylinder engine; mean height of the curves increased with engine speed. Nearly perfect resonance effect at speed of 1240 r.p.m. is noticeable.

In this test the exhaust pipe was about 9 feet in length, and was provided with two simple expansion boxes of about 1 cu.ft. capacity each placed in series near the outlet end. These particulars are given because the design of the silencer and the dimensions of the exhaust piping have a large influence on the pressures. By experimenting with various sizes and lengths of exhaust pipes, it is frequently possible to improve the volumetric efficiency and the power output of the engine.

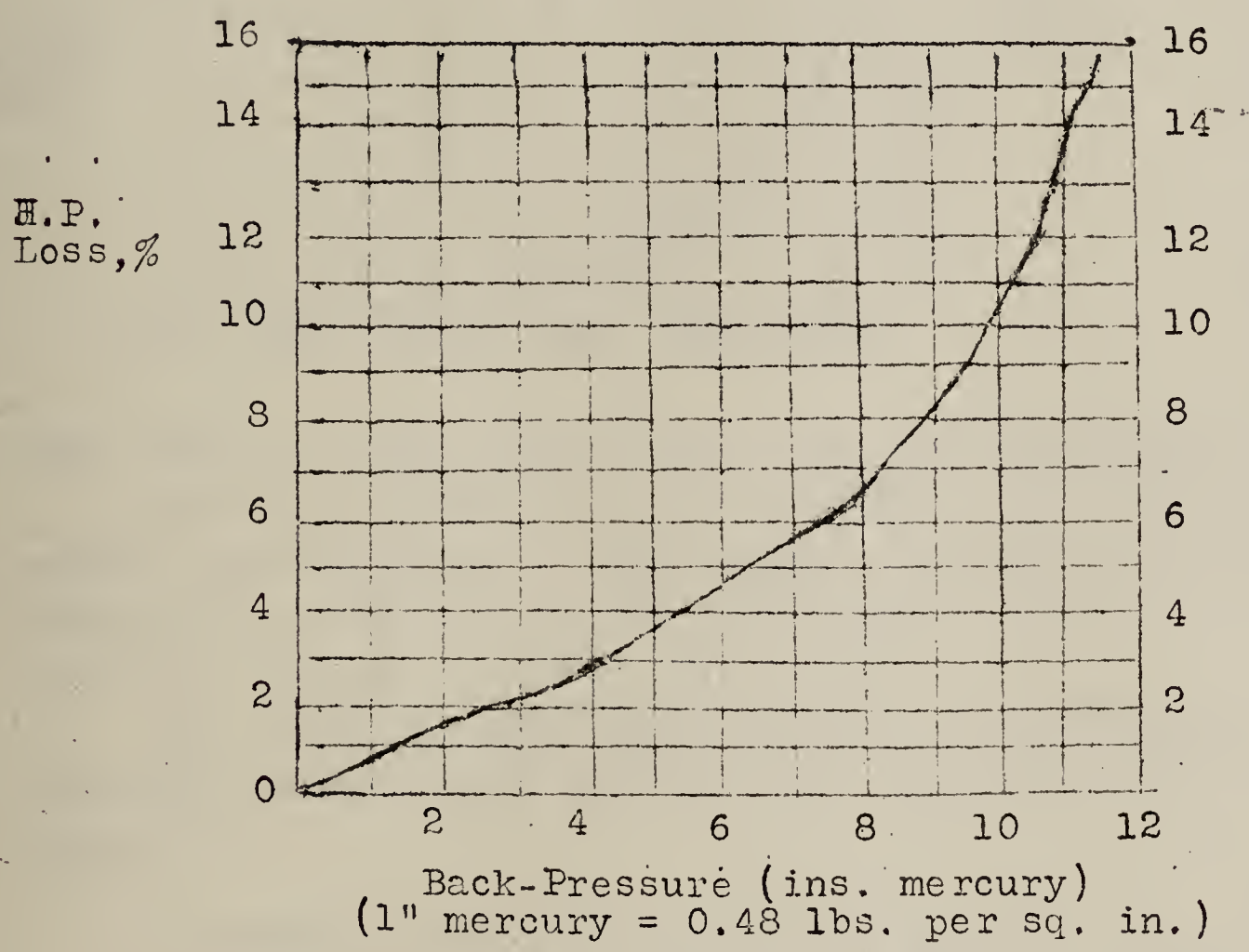
Effect of Exhaust Pressure on Power.

Influence of exhaust back-pressure on power output is of much importance, and is a question which is involved in the design of the exhaust manifold and silencer. With exhaust pipe of adequate dimensions, free from bends and all constrictions, the engine will give its maximum power. If the flow of exhaust gases is restricted, the exhaust back-pressure will rise, and its effect is to raise the exhaust line of the diagram, which reduces its area and the power output. If the throttling effect is small, the reduction in power will be proportional to the increase of the back pressure; but as the throttling effect increases the reduction in output will be proportional to a higher power of the back pressure. Also, throttling the exhaust-gas flow increases the temperature of the gases and of the parts in contact with them including cylinder head and walls,- a falling-off in the volumetric efficiency. Confined exhaust gases tend to expand into the cylinder. If restrictions and throttling effects are excessive, the combined effect of pressure rise, temperature increase, and exhaust-gas expansion may stop the engine.

Practically all silencing devices introduce some resistance, and usually result in increase of back-pressure and loss of power,- the greater the degree of silencing the higher the back-pressure and the greater the power loss.

Results of Back-Pressure Tests made by the U.S. Advisory Committee for Aeronautics on an eight-cylinder Curtiss engine, cylinders 4" bore by 5" stroke; engine rating, 70 H.P. at 1200 rpm.

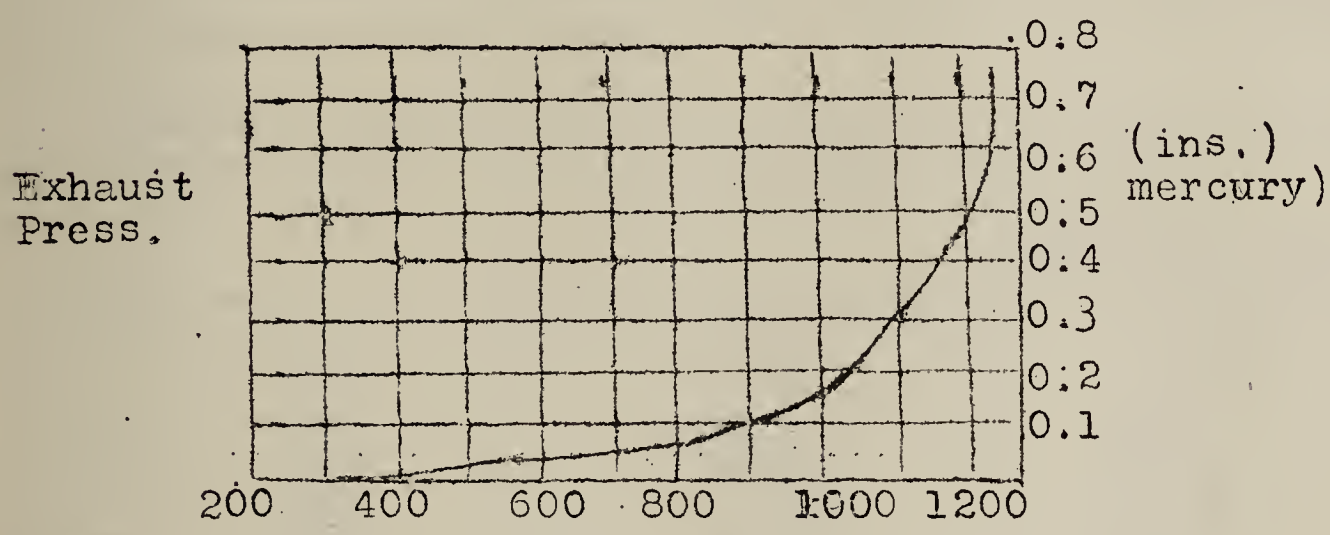
Curve showing relation between exhaust back-pressure and power loss.



Up to about 5 or 6 inches of mercury back-pressure, power loss is practically proportional to the back-pressure, thereafter the power loss increases at a more rapid rate. At a back-pressure of 12 ins. of mercury the loss is about 1/5 the total putput. With an open exhaust, at 1230 r.p.m. engine gave about 65 B.H.P. A back-pressure of less than 10 lbs. per sq. in. would stop the engine.

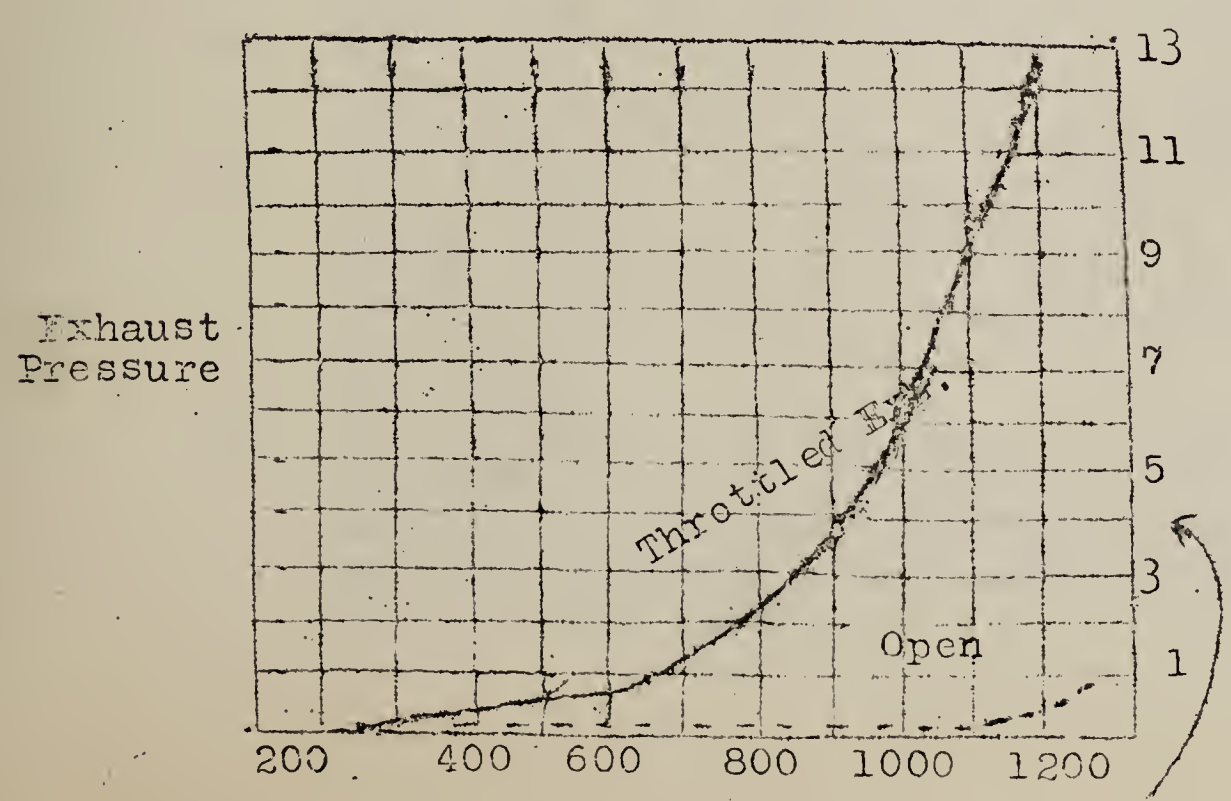
Results show that for back-pressures up to 2 or 3 lbs. per sq. in. above the normal open exhaust back-pressures, the loss in brake M.E.P. does not exceed, but is nearly equal to, the back-pressure. The curve shows that the back-pressure varies as the 1.5 power of the B.H.P.

Variation of Exhaust Back-Pressure with Engine Speed is shown in curve of results of tests on same Curtiss engine. Curve shows that, with the usual exhaust manifold and connections, the back-pressure rises rapidly at the highest range of speeds. The maximum back-pressure, nevertheless, was less than 0.5 lbs. per sq. in. with an open exhaust pipe.



With an ordinary silencer attached, the back-pressure at the normal working speed of 1200 r.p.m. would be about 1.5 to 2.5 lbs. per sq. in.; and for a very efficiency silencer, the back-pressure would be from 1 to 2 lbs. per sq. in.

The corresponding back-pressures at different engine speeds with the exhaust pipe throttled to a nearly closed position is shown.



It was noticed that at certain speeds there was an abnormal drop in the power loss, andthat this effect ceased for small changes o speed on either side of this critical speed. Also, that by changing the exhaust manifold, the critical speed eith

changed. The abnormal increase of power loss was probably due to reflected wave of the exhaust gas filling the combustion chamber of one or more cylinders just before the exhaust valve closed.

Effect of Exhaust Pipes. The power output of an engine exhausting straight into the atmosphere is generally less than when lengths of exhaust pipes are used, and an open exhaust pipe gives more power than the best designed silencer.

Results of exhaust-pipe tests on a four-cylinder engine of 4" bore by 5.5" stroke.

Engine R.P.M.	With Silencer	Without Silencer or Exhaust Pipe	With Exhaust Pipe only
	H.P.	H.P.	H.P.
500	14.0	14.25	14.25
750	22.5	23.5	23.5
1000	28.5	30.25	30.5
1250	33.25	36.5	37.25
1500	36.0	40.5	42.0
1750	36.75	41.75	44.5

Results show that the power output with the exhaust pipes only was more at all speeds than without the exhaust pipe or with the silencer attached.

Silencer Requirements. The principles underlying correct design of silencers may be briefly stated as follows:

1. A good silencer should give the maximum reduction of noise with the least relative loss of power.
2. The most efficient silencers are those in which the volumetric capacity of the silencing chamber is not less than 0.008 cu.ft. per horsepower.
3. The principle of cooling the gases by causing them to sweep over a relatively large area of silencer surface which is exposed to the natural air draught due to the motion of the engine through the air appears to give good results,- reduction of volume of the exhaust gases by cooling.
4. The expansion of the gases from the combustion-chamber pressure down to atmospheric pressure should be as gradual as possible, and the design of the silencer should aim at a gradually increasing volume from the manifold to the final outlet.

5. The use of baffle plates and gauzes should be avoided as far as possible.
6. A well-designed silencer, complete with the exhaust pipes (not including the manifold) should not weigh more than about 1/4 pound per B.H.P.

Results of test on automobile silencers by Judge showed that it was possible to obtain a marked reduction in exhaust noise with a power loss varying from 1% at two-thirds full speed up to about 5% at highest engine speed. For a better silencing effect (such as is used in best automobile practice) there was a power loss varying from about 1.5% at two-thirds full-speed up to about 7% at highest engine speed.

The most efficient silencers seem to be those of comparatively simple construction with liberal channels for the flow of the gases. With baffles, perforated plates and gauzes, periodical cleaning has to be considered.

Results of Measurements

Aim to reduce the engine losses at all speeds to a minimum.

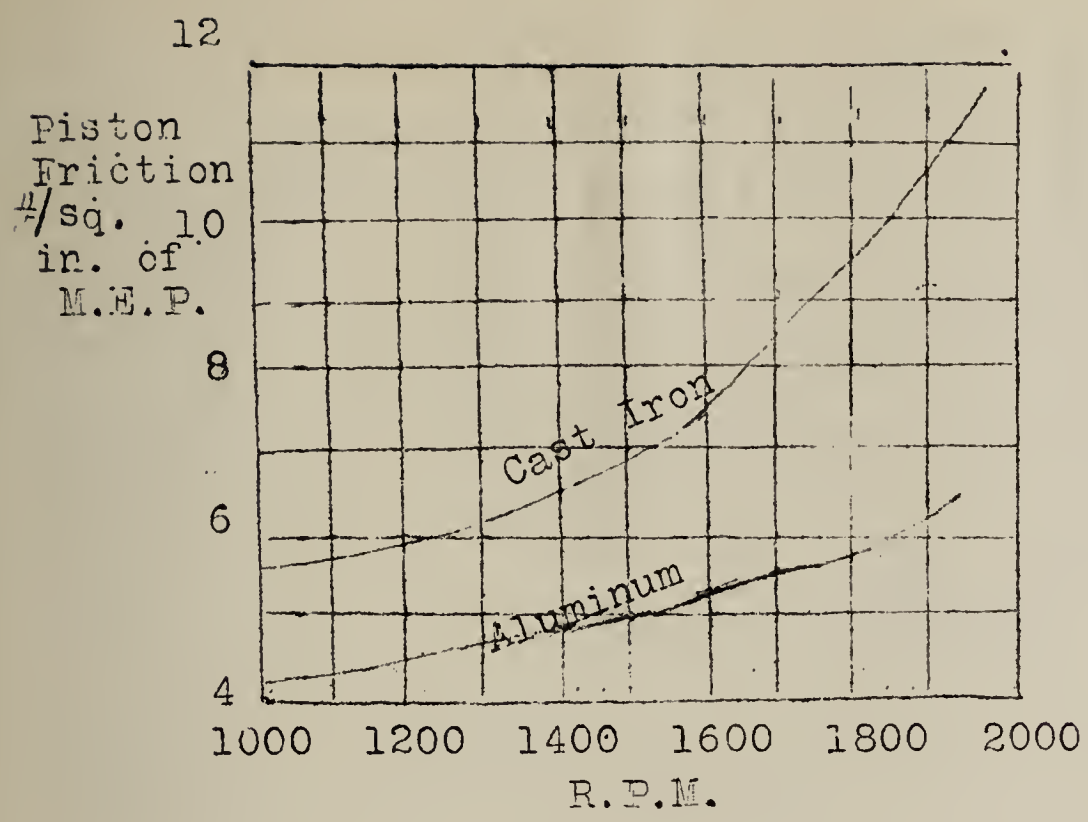
Engine-power Losses. Relative value of Friction Losses and Pumping Losses.

In tests on a Talbot early type four-cylinder automobile engine the friction losses were about 75% of the whole loss, while the pumping losses were about 25%. In tests on more recen t Daimler engine, the friction losses were 82% and the pumping losses 18% of the whole. With modern engines of high volumetric efficiency, the pumping losses should not exceed 12% to 15% of the whole loss; a fair value is 12%.

Frictional Losses. Besides the frictional resistance offered to motion by the various working parts, there are other losses such as churning losses in the crank case, valve-gear losses, vibrational and out-of-balance effects, etc. The power used-up in driving the auxiliaries, the magneto or generator, the water circulating pump, the oil pump, etc , is included in the mechanical so-called frictional losses of the engine as a whole. All these losses depend somewhat on the design of the particular engine, including the weight, the material used, means of lubrication, etc.

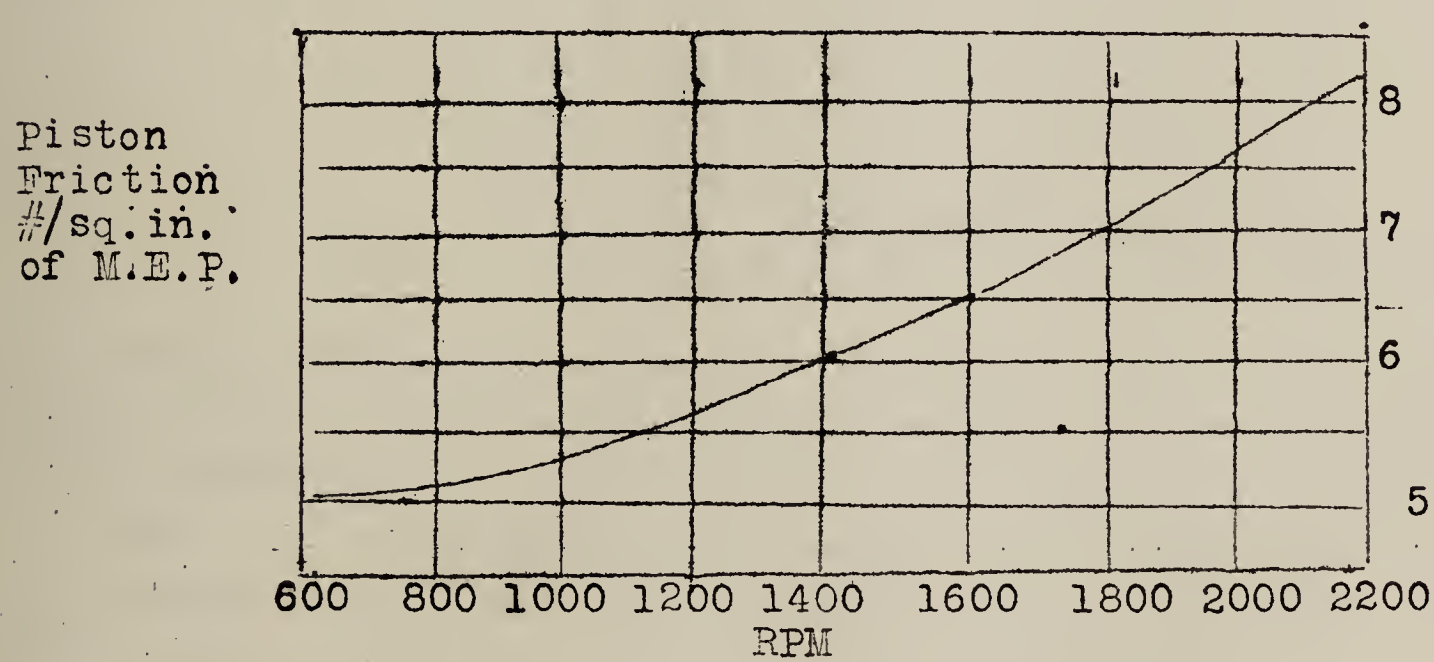
Piston Friction. Piston friction is the most important of all frictional losses. Rough estimate of the amount of piston friction expressed as horse-power may be gained by methods that are approximate; difficult to arrive at conclusive results by analytical methods.

As a result of a large number of tests made to determine both the cause and the extent of piston friction, Ricardo found that the piston friction was much greater than that of an equivalent bearing surface in other parts of the engine on account of the fact that the piston's motion is reciprocating and not continuous, and that the lubricating oil is always more or less contaminated and carbonized and of reduced viscosity.



The curve shows the comparative piston frictions of a cast-iron and an aluminum "slipper" piston.

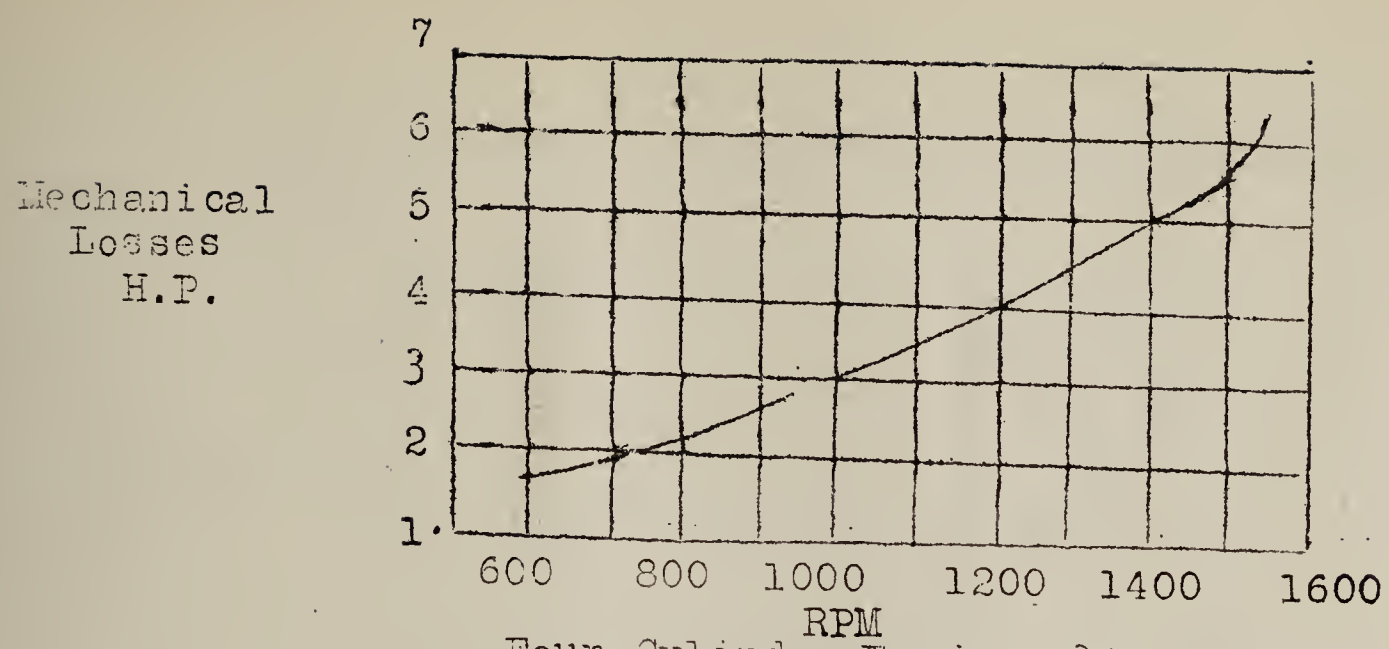
Results of Ricardo's tests on a gasoline engine of 7 1/4" bore by 8 1/2" stroke, piston speed 1980 ft. per min., are expressed in the curve.



This curve shows the loss due to piston friction alone when working over its appropriate range of speed.

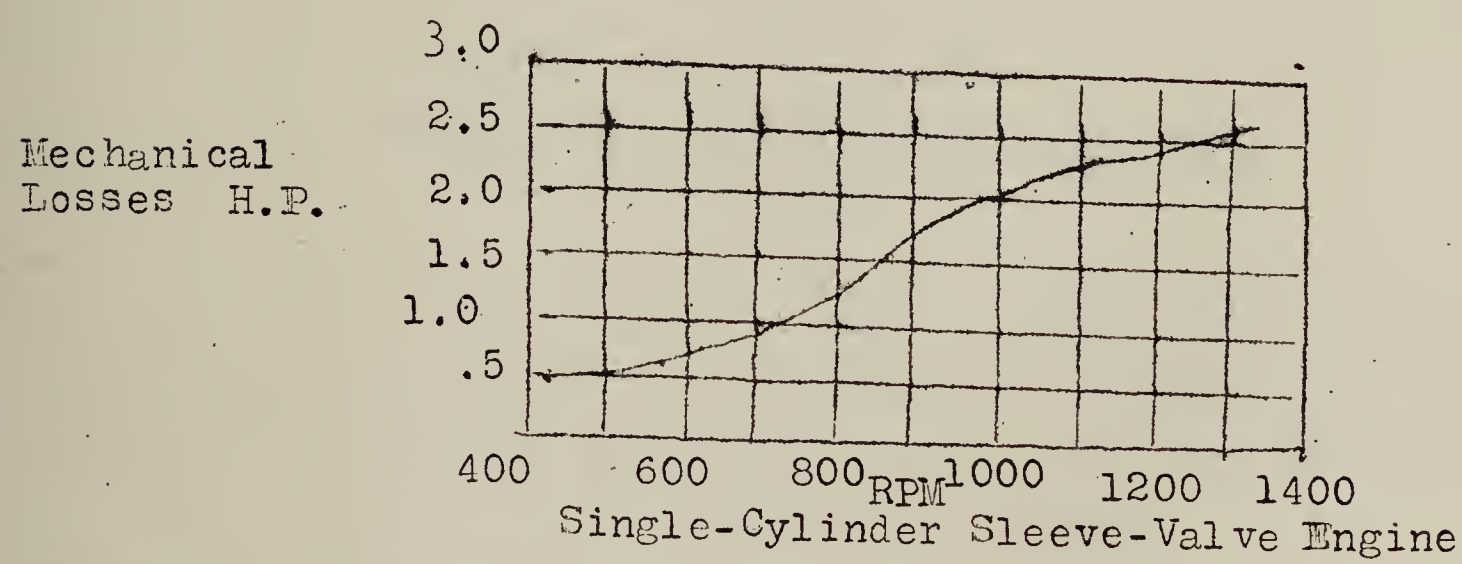
<u>Mechanical Losses</u>. The effect of speed on friction losses is to cause an increase in the losses.

Curve representing the losses at different speeds in a four-cylinder poppet-valve automobile engine.



Four-Cylinder Engine, 30 H.P.

Curve representing the losses at different speeds in a single-cylinder sleeve-valve engine.



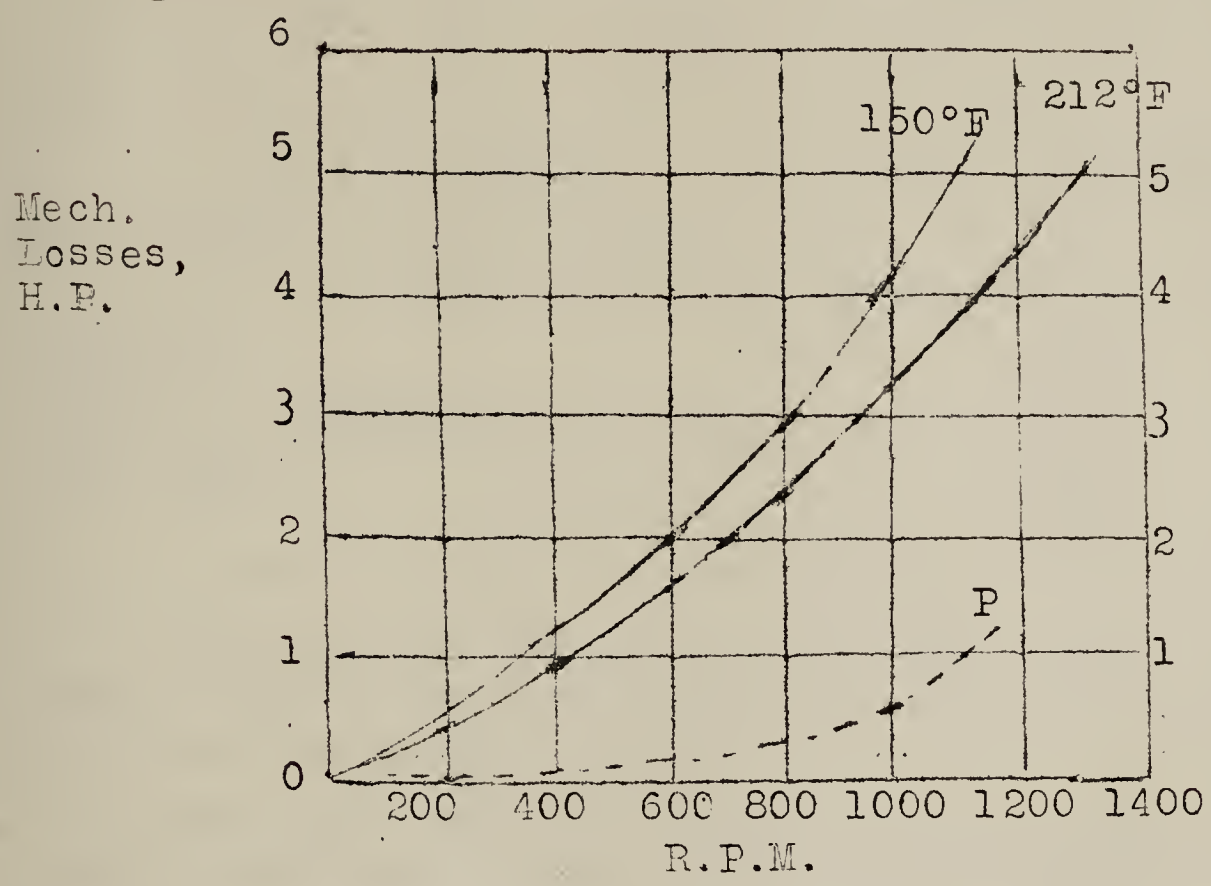
Single-Cylinder Sleeve-Valve Engine

It will be seen in the two cases that the mechanical losses do not vary directly as the engine speed, but as some higher power of the speed. This power index is approximately 1.5 to 2.

The two curves should not be used as a comparison of mechanical efficiency because the total power of the two engines is quite different.

It should be mentioned that the mechanical losses represented in these two curves include the pumping losses.

Results of tests by Hopkinson on a Daimler engine are here given.



Curves show the total losses for two different temperatures of the cooling water. The dotted curve, P, represents the pumping losses; so, the difference between the ordinates of this curve and the other two curves gives the mechanical frictional losses at the corresponding speeds.

Analysis of Frictional Losses.

Friction loss is made-up of several parts.

Friction losses depend on,-

1. Frictional losses due to the static pressure of the exploding gases (and compressing).
2. Frictional losses due to rotating parts.

Losses (1), remain constant with speed, unless mean effective pressure drops with speed, then these losses will be slightly lower.

Losses due to rotating parts depend on the pressures between these parts; losses will go-up in proportion to speed.

Power required to drive the auxiliary pumps increase at a greater rate than with the speed.

Hopkinson found that it took 4.0 H.P. to drive a certain engine 180 r.p.m. with the valve cape removed, that about 2.6 H.P., or 65% of the whole friction loss, was due to the piston and

crank-pin friction, and that about 1.4 H.P., or 35% of the whole friction loss, was due to the main bearings, valve-gear and minor sources.

Results of measurements of the power to drive the various auxiliaries as made by Judge are given in the table.

Part of Engine	H.P. to Drive	Per-cent of whole
Whole engine (caps removed)	8.8	
Magneto	0.25	2.8
Circulating pump	0.25	2.8
Pump and magneto skew gearing	0.40	4.5
Air Pump	0.30	3.4
	1.20	13.6

The speed of the engine was 1000 r.p.m.; and the temperature of the circulating water and lubrication were kept constant at about the working values.

The power absorbed by the auxiliary parts represents about 14% of the total mechanical friction losses.

The power required to drive a magneto for a two-cylinder engine at different speeds is shown in the table. In this case the magneto speeds are one-half the engine speeds.

Magneto Speed in R.P.M.	Horse-power to Drive
700	0.0269
800	0.0362
1,000	0.0491
1,200	0.0668
1,400	0.0695
1,600	0.0794
1,800	0.0892
2,000	0.1000

Ricardo gives the following values for a 100 H.P. six-cylinder engine.

	lbs. per sq. in. pressure
Bearing Friction	0.75 to 1.00
Valve Gear	0.75 to 0.80
Magneto	0.05 to 0.10
Oil Pump	0.15 to 0.25
Water Pump	0.30 to 0.50
Total	2.00 to 2.65

Results show that the frictional losses, exclusive of piston friction, vary from about 1.5 lbs. per sq. in. of mean effective pressure for modern six, eight, or twelve cylinder aero-engines up to over 3 lbs. per sq. in. for large heavy gas engines. A good value for the average high efficiency aircraft or automobile engine is from 2. to 2.5 lbs. per sq. in. of mean

effective pressure.

Effect of Temperature of Circulating Water. Some of the frictional losses vary with the cylinder-wall temperature.

Hopkinson found from experiments on a Daimler engine that with the circulating water at a temperature of 65° F. the frictional losses were 4.0 H.P. While at 150° F. they were 2.6 H.P. and at 212°F. they were only 2.0 H.P. The speed in each case was 800 r.p.m. and the total output was 16 H.P.

Dr. Watson found from tests on a sleeve-valve engine that the B.H.P. increased rapidly as the temperature of the entering circulating water rose: At 70° F., the B.H.P. was 4.6, while at 180° F. it became 5.95. The I.H.P. was 8.1 and remained constant over the range of temperature.

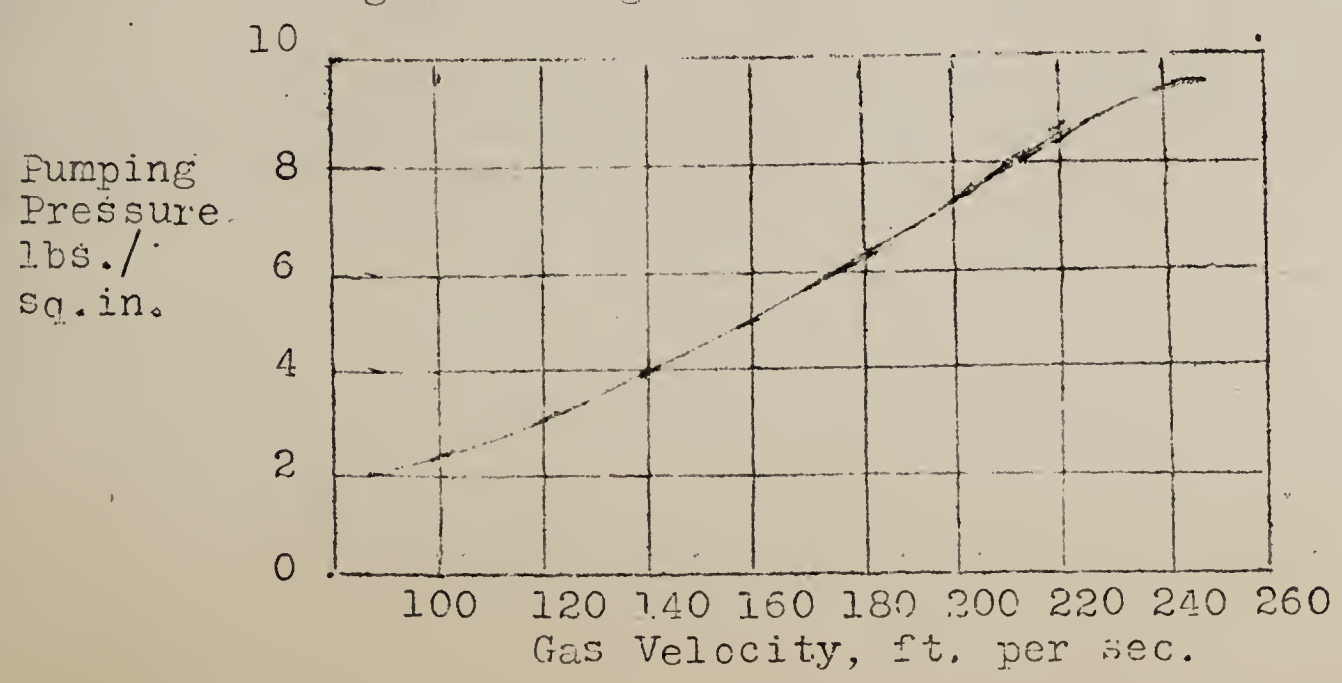
These examples serve to show that the temperature of the lubricant has an important influence on the frictional resistance.

The Pumping Losses.

The pumping losses include the work done by the engine in inducing the charge into the cylinder and expelling the exhaust products from it. The "negative work," which has to be performed in the two strokes is represented by the smaller loop of the indicator diagram. This negative work is usually between 10% and 20% of the total losses between the indicated and brake horsepower of an automobile engine.

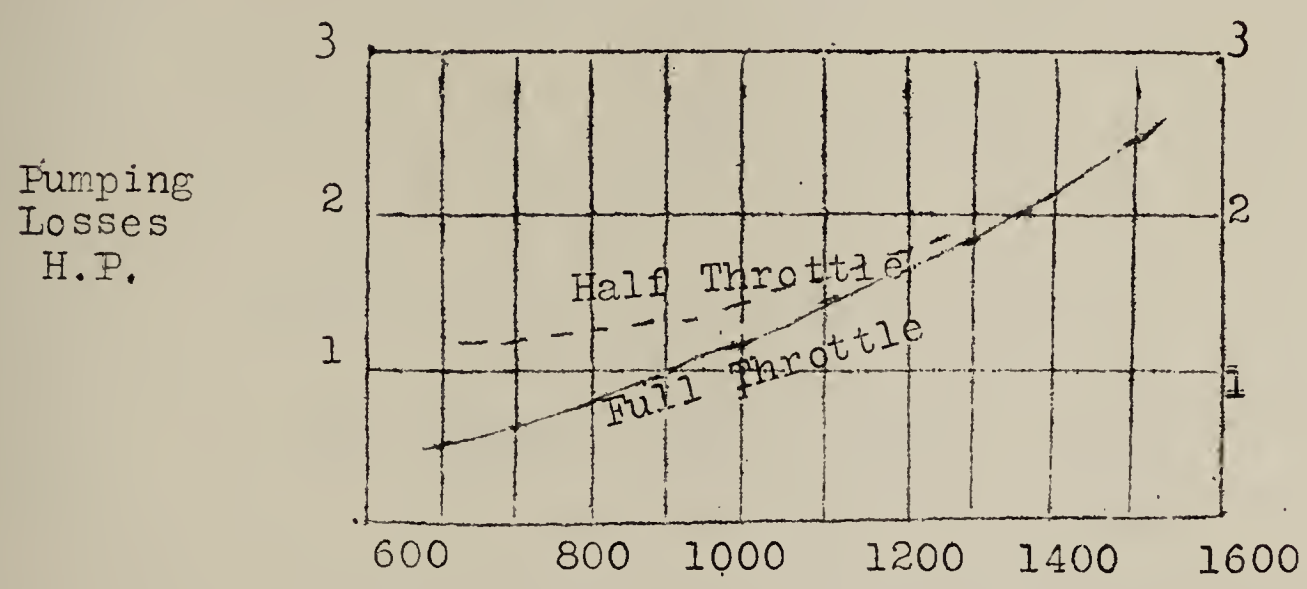
The pumping losses depend on the inlet and exhaust valve lifts and timing, the design of the induction and the exhaust ports and passages, the design of the carburetor and silencer, etc. With well-designed induction passages of fairly short length. The loss due to pumping effects may be assumed to be proportional to the velocity of flow of the gases through the valves.

The curve represents the pumping losses at different gas velocities in good average automobile and aircraft engine practice.



It has been found that the temperature existing within the cylinder has an influence on the loss.

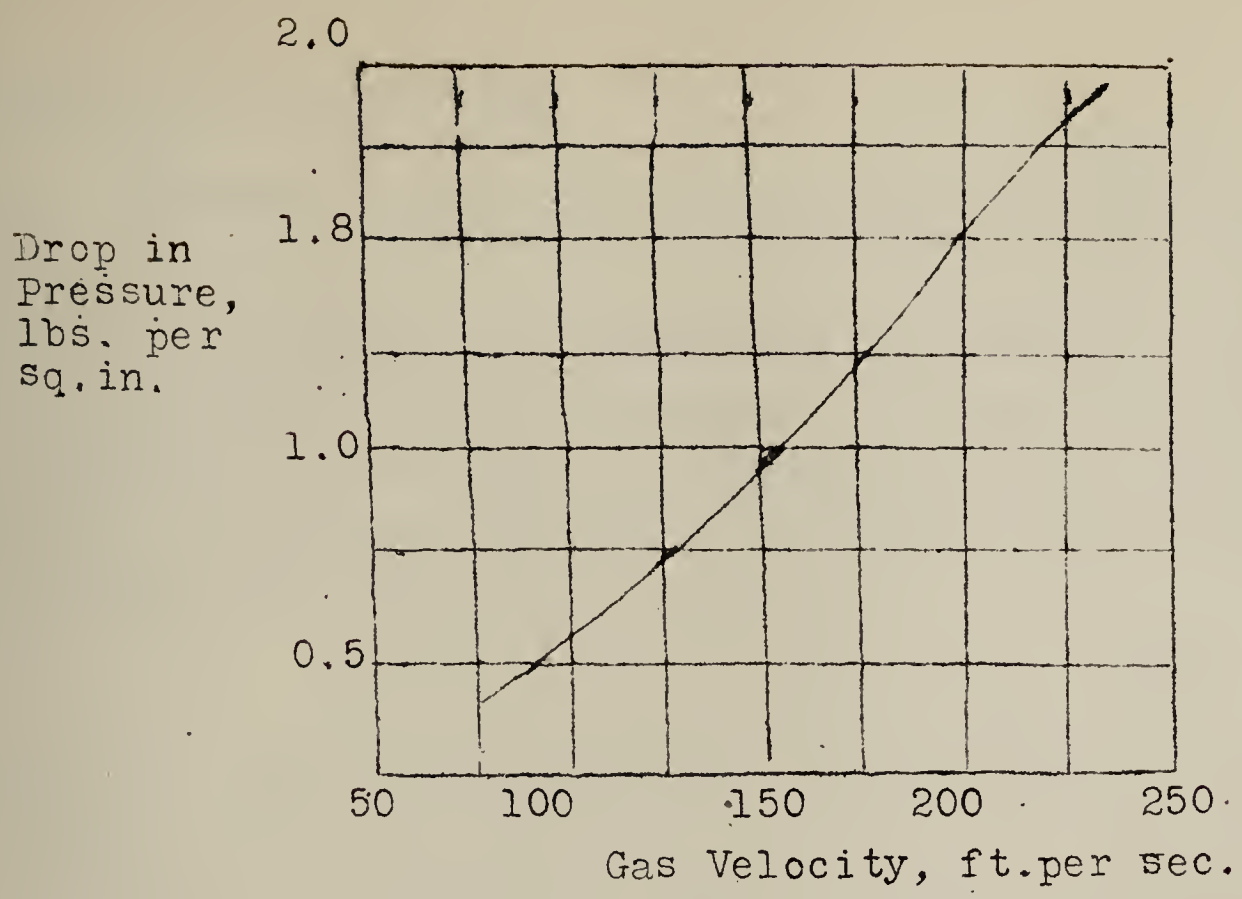
The pumping losses in a four-cylinder engine at different speeds as obtained from measurements of indicator diagrams are shown in the curve.



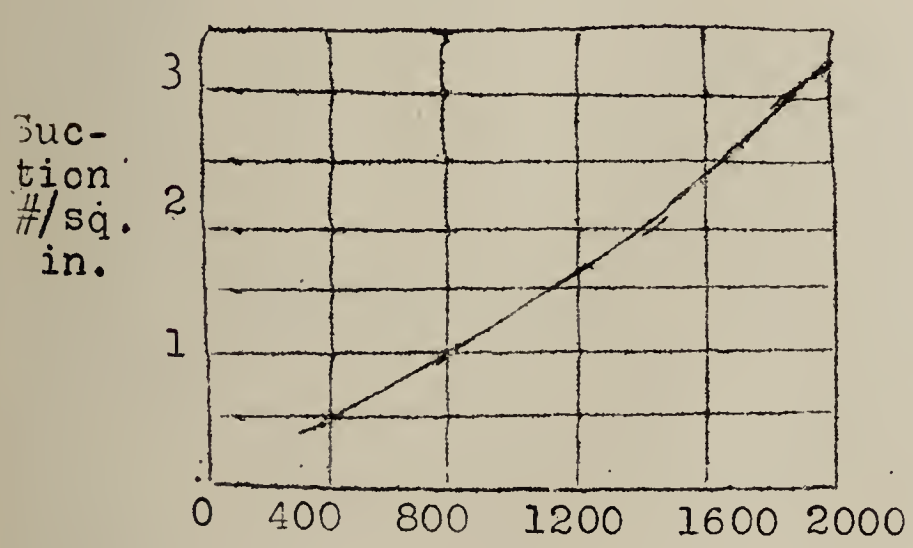
Drop in Pressure during Suction Stroke. Effect of piston suction in drawing the charge through the inlet valve is to cause a drop in pressure in the cylinder due to the increased velocity of the mixture through the valves. In good engine practice, the velocity of the charge through the valves is never less than 100 ft. per sec. in order to obtain proper turbulence and combustion. For a gas velocity of 100 ft. per sec., the pressure drop may vary from 0.3 to 0.4 lbs. per sq. in. for no induction pipe nor carburetor.

In order to reduce the pressure drop to a minimum, the valve ports should be ample, the induction pipe short and as free as possible from bends. The use of short smooth induction pipes and overhead valves of proper area, may reduce the pressure drop through the inlet pipe and valves to 1/4 lbs. per sq. in. The area of the suction loop will depend in part on this pressure drop.

The total pressure drop at different gas velocities for a good modern engine practice, due to all contributing influences including the carburetor is shown in the curve.



It is seen that the losses increase with increased gas velocity. This also follows from consideration of the average pressures existing in the inlet and exhaust pipes. The results of measurements of the negative pressures in the inlet manifold at different speeds with a four-cylinder automobile engine are shown in the curve.



The values are higher than would be anticipated in a modern engine, but they serve to show, as before, that the suction pressure increased fairly rapidly with speed. In the case of the exhaust gases, the back-pressure also increases with speed in a similar manner.

The difference between the inlet and exhaust pressures increases very appreciably as the engine speed increases, and therefore, the loop of negative work would be larger at the higher speeds,- increase not as great as might be expected.

Mechanical Efficiency

$$\text{Mechanical Efficiency} = \frac{\text{B.H.P.}}{(\text{B.H.P.}) + (\text{Total Engine losses})}$$

From theoretical considerations based on results of tests on different engines, Professor Callendar concludes that the mechanical efficiency is a function of the diameter of the cylinders,- the larger the size, the higher the efficiency. He expresses the relation as follows:

$$\text{Mechanical Efficiency} = K\left(1 - \frac{0.3}{D}\right)$$

K is a constant which varies with different engine designs. Callender proposes a value for K of 0.93. It is certain that mechanical efficiencies will be higher for modern engines.

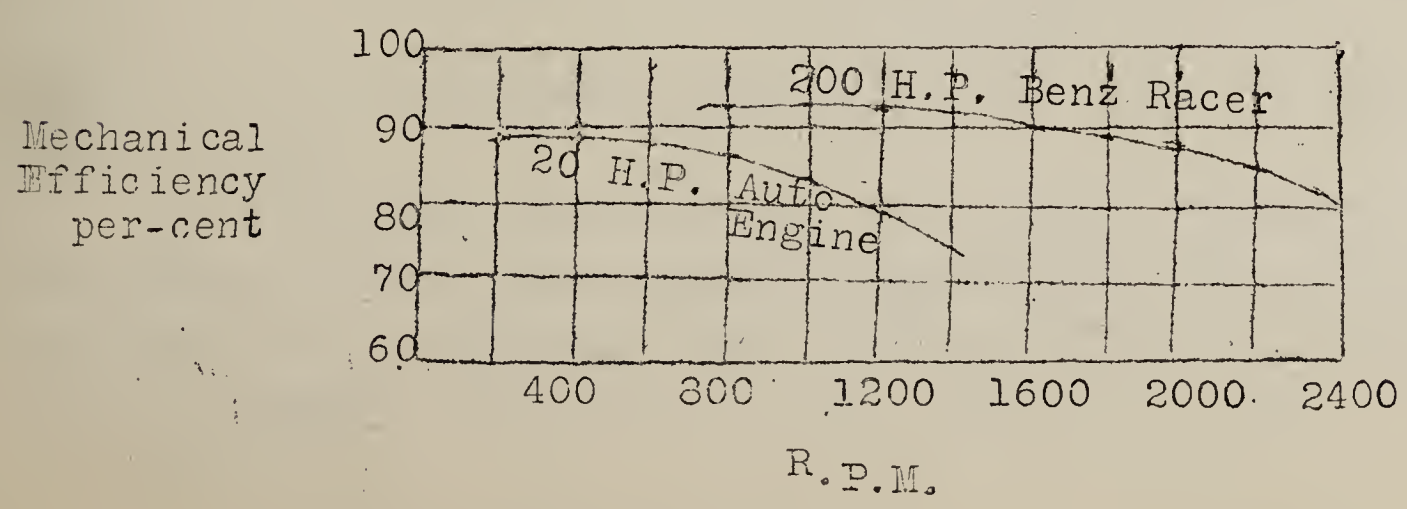
A modern light automobile engine 2-5/8" bore by 4" stroke gave a mechanical efficiency of 87.5% at 1600 r.p.m. and 89% at 1000 r.p.m. An engine of 7" bore by 8" stroke gave mechanical efficiencies of 90% and 93% at 1600 and 1000 r.p.m. respectively.

A better value for the coefficient, K, would be from 0.97 to 0.98 for 1000 r.p.m.

The mechanical efficiency of a modern motor-car engine varies from about 87% to 92% for speeds from 1000 to 1500 r,p.m. and falls-off to about 78% to 84% for the higher speeds from 2500 to 3000 r.p.m. Much depends on the actual design of the engine.

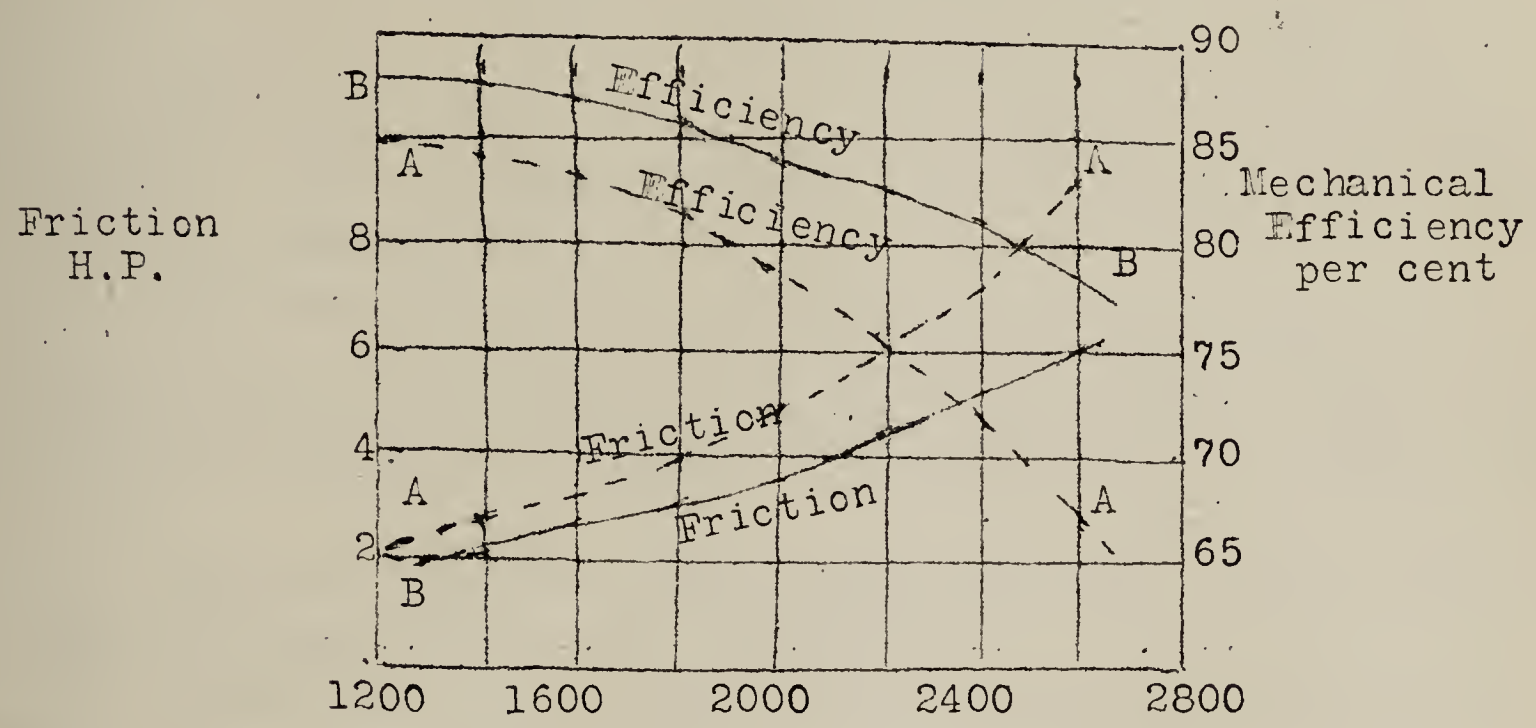
Effect of Speed. Since mechanical losses increase with engine speed at a higher rate than the speed, the mechanical efficiency will progressively fall as the engine speed increases.

The following curves show the mechanical efficiencies of a touring-car engine of an early type, and of a racing engine at different speeds.



Efficiencies fall-off as the speed increases. The results obtained with the racing engine represent more nearly the case of modern engines.

Results of tests made by Ricardo on an L-head engine, 2-3/4" bore by 4" stroke.



Curve A is for the engine with cast-iron pistons. Curve B is for the same engine with aluminum slipper pistons and a newly designed cylinder head giving greater turbulence and compression.

Horse Power

Factors which are responsible for the indicated horsepower are the fuel or mixture, and the design of the combustion chamber; also, volumetric and thermal efficiencies of the engine.

The I.H.P. is readily determined from knowledge of the mean effective pressure and the engine speed, these are the only variable quantities in the rational formula

$$\text{I.H.P.} = \frac{(\text{m.e.p.})\frac{\pi}{4}D^2 \times \frac{S}{12} \times \frac{\text{r.p.m.}}{2}}{33{,}000} \times M$$

where M is the number of cylinders in a multi-cylinder four-stroke cycle engine.

It therefore follows that, for a given cylinder size, the horsepower can be increased only by working the engine at higher mean effective pressures and at higher number of revolutions. Progress in power-output has been due largely to increase of these two factors.

The manner in which the B.H.P. varies with engine speed in typical airplane engines is shown in the curves on page 84.

The corresponding torque-speed curves for these engines were estimated from the relation

$$\text{Torque} = 5250 \frac{\text{B.H.P.}}{\text{r.p.m.}}$$

The torque curves are useful in connection with propeller design. Maximum engine torques occur at rather lower speeds than for maximum B.H.P.

The dimensions of these engines are as follows:

A. Liberty 400. 12-cyl. 5" bore by 7" stroke, 1700 r.p.m.
B. Napier Lion 400. 12-cyl. 5" bore by 5-1/8" " , 2000 r.p.m.
C. Rolls-Royce 300 12-cyl. 4-1/2" bore by 6-1/2" stroke, 1800 r.p.m.

Typical power and torque curves are also given for motor car and light car engines (Page 85).

These power curves have same general characteristics as those for the airplane engines.

The dimensions of these engines are as follows:

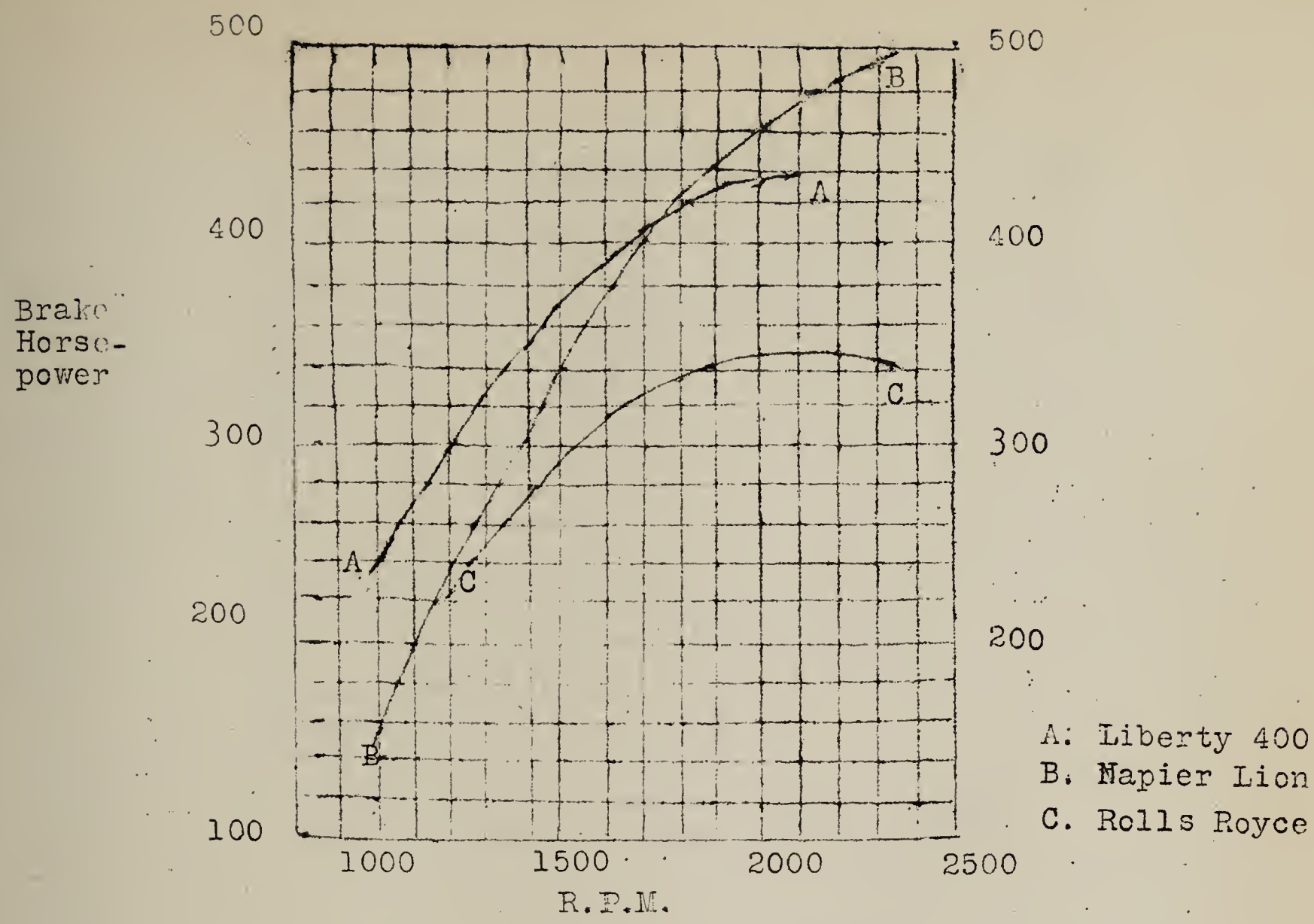
A. Calthorpe racer 4-cyl. 3-1/8" bore by 5-7/8" stroke.
B. Talbot Darracq 8-cyl.
C. Moline-Knight 4-cyl. 4" bore by 6" stroke
D. Pomeroy 4-cyl. 3-1/2" bore by 4-5/8" stroke
Morris Oxford
E. Aluminum Pistons } 4-cyl. 2-3/4" bore by 4" stroke
F. Cast Iron Pistons }

<u>Brake Mean Effective Pressure</u>. Brake m.e.p. is more widely used in engine comparisons than the indicated m.e.p. Measured B.H.P. only requires to be divided by the r.p.m. and a constant for the engine size to obtain the brake m.e.p.

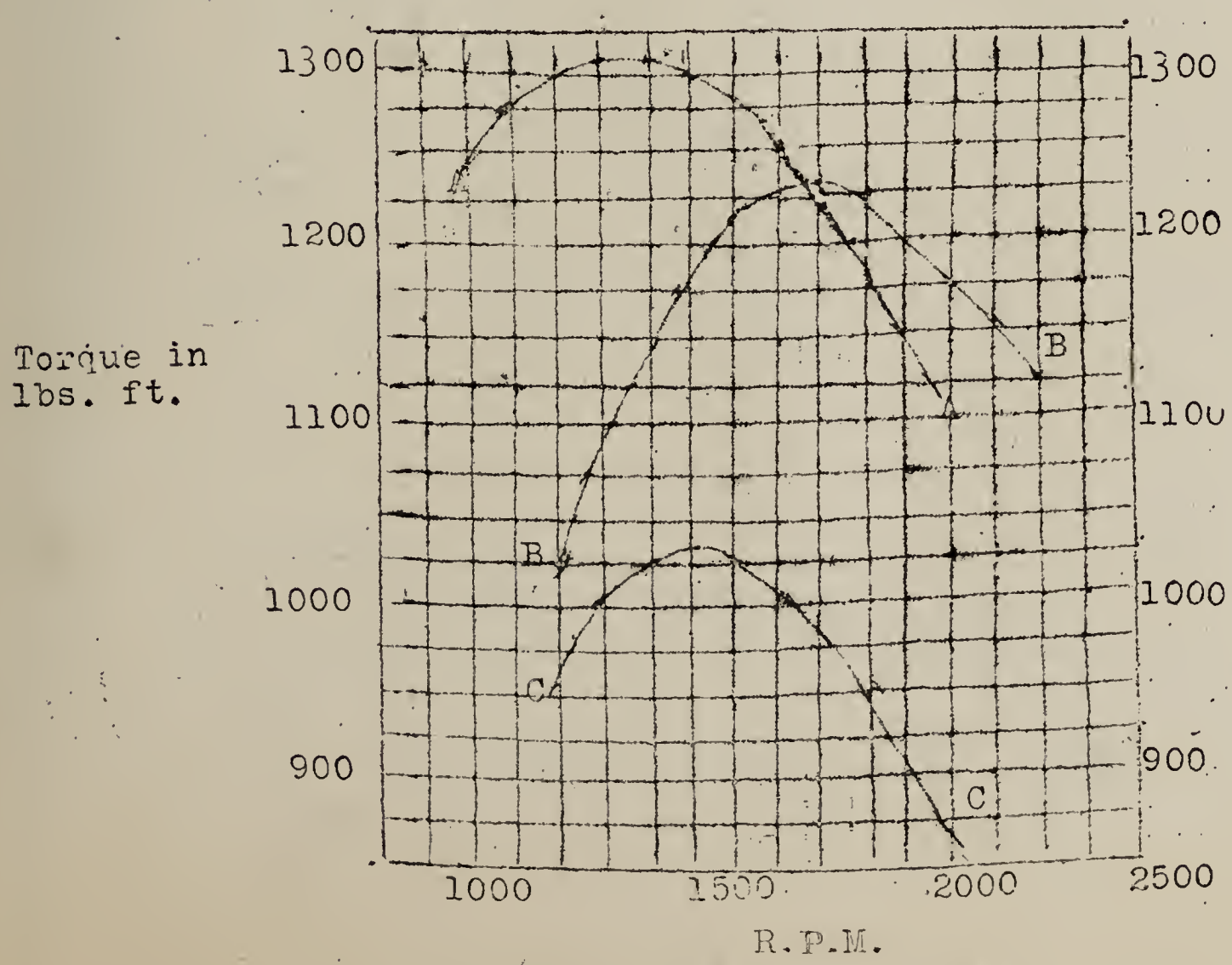
Useful form for this relation,-

$$\text{Brake m.e.p} = 792{,}000 \frac{\text{B.H.P.}}{\frac{\pi}{4} D^2 (\text{r.p.m.}) M}$$

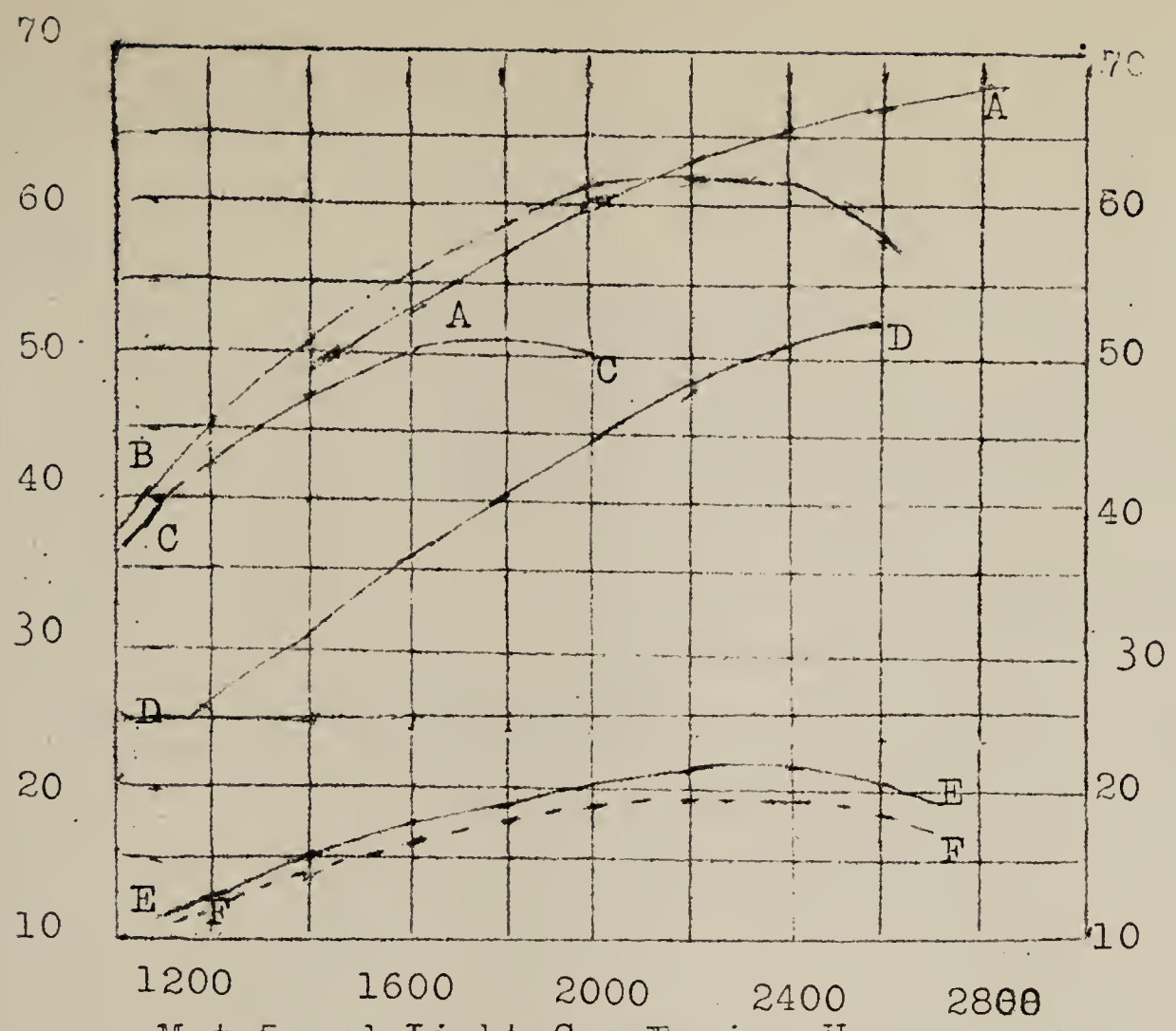
Brake m.e.p.= Indicated m.e.p. X mechanical efficiency



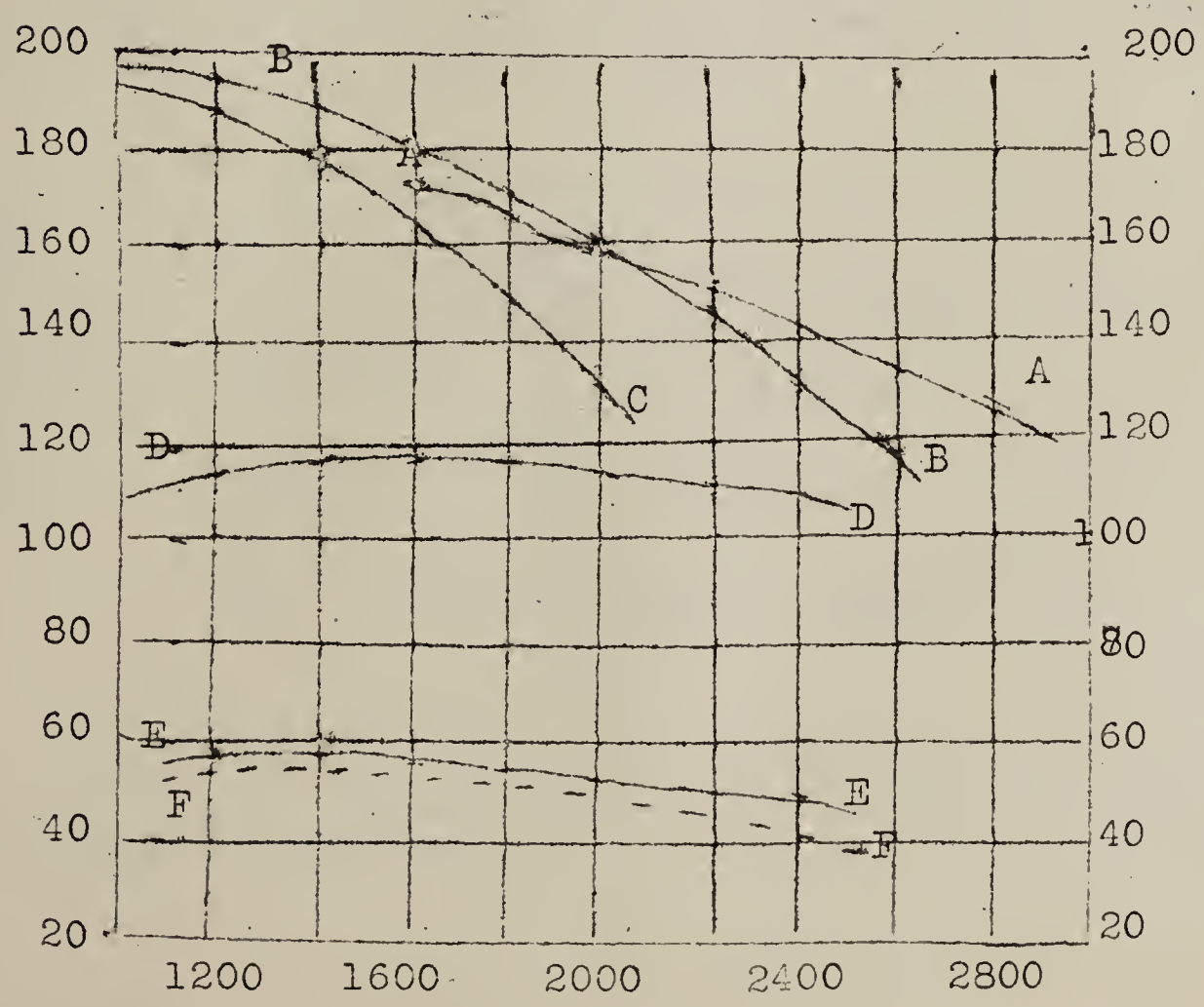
Airplane Engine Horse Power



Airplane Engine Torque



Motor and Light Car Engine Horsepower



Motor and Light Car Engine Torque

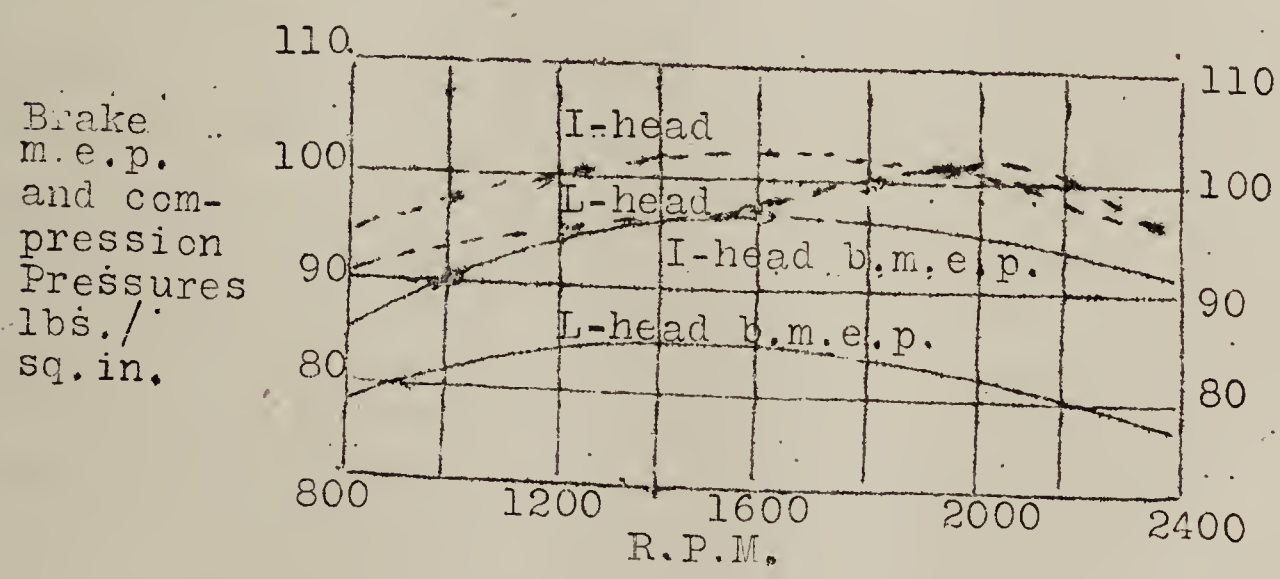
Factors Influencing Brake m.e.p. Brake m.e.p. is not constant for any or every engine but varies between about 70 and 130 lbs. per sq. in. Its actual value is governed by the following factors:

1. Compression Ratio
2. Engine Speed
3. Mixture Strength
4. Shape of Combustion Chamber
5. Dimensions of the Engine
6. Volumetric Efficiency

Also varies to a certain extent, with temperature and density of admitted air, and temperature of cooling water.

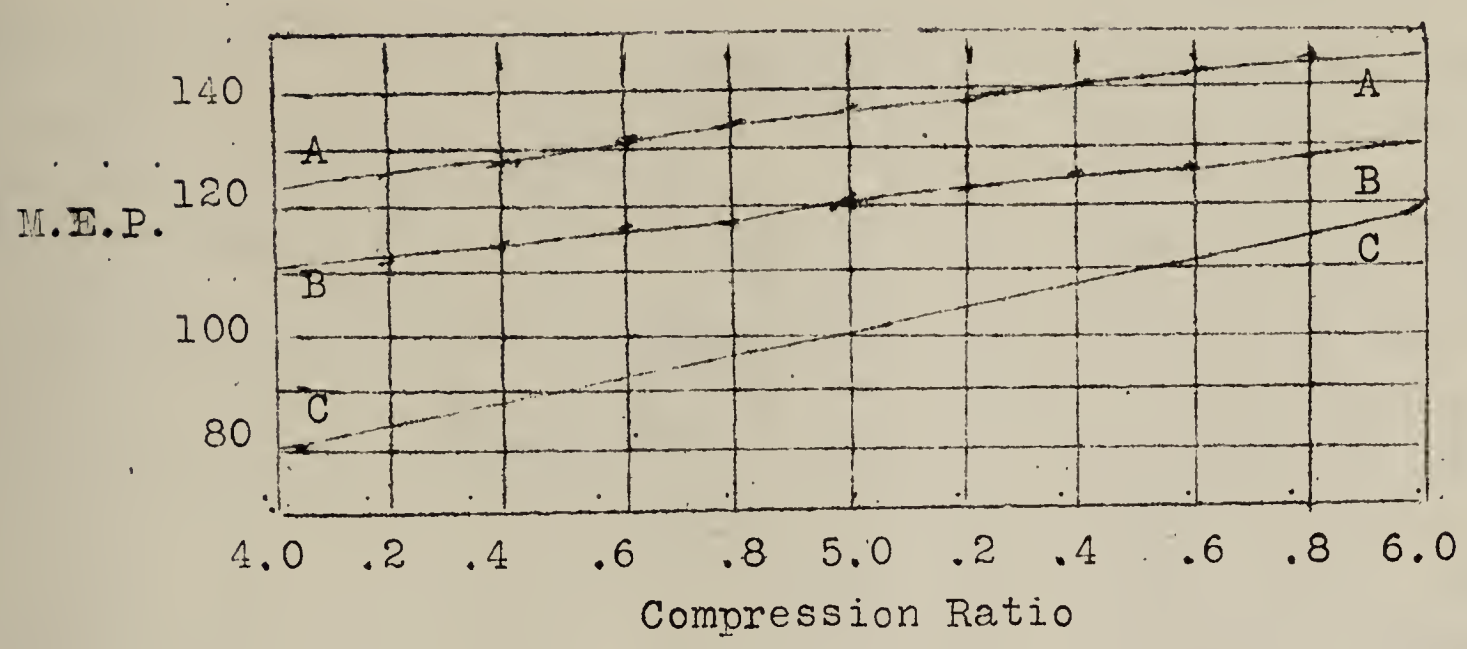
Effect of Compression Ratio General effect of increasing the compression ratio is to increase the area of the indicator diagram, and the mean effective pressure. For usual compression ratios in motor-car engines of 3.5 to 5.5, the corresponding values of brake m.e.p. would be about 65 to 125 lbs. per sq. inch.

Curve showing relation between compression pressures and brake men effective pressures at different speeds in the case of an L-head and a valve-in-head four-cylinder motor-car engine. The two engines had same compression ratio, 4.8 and were nearly the same size, bore 3-1/2", stroke of L-head, 4-3/4", stroke of I-head, 4-5/8"+.



The difference in the values of the brake m.e.p. for the two engines is chiefly due to the difference in design of the combustion chamber.

Curve shows relation between m.e.p. and compression ratios.



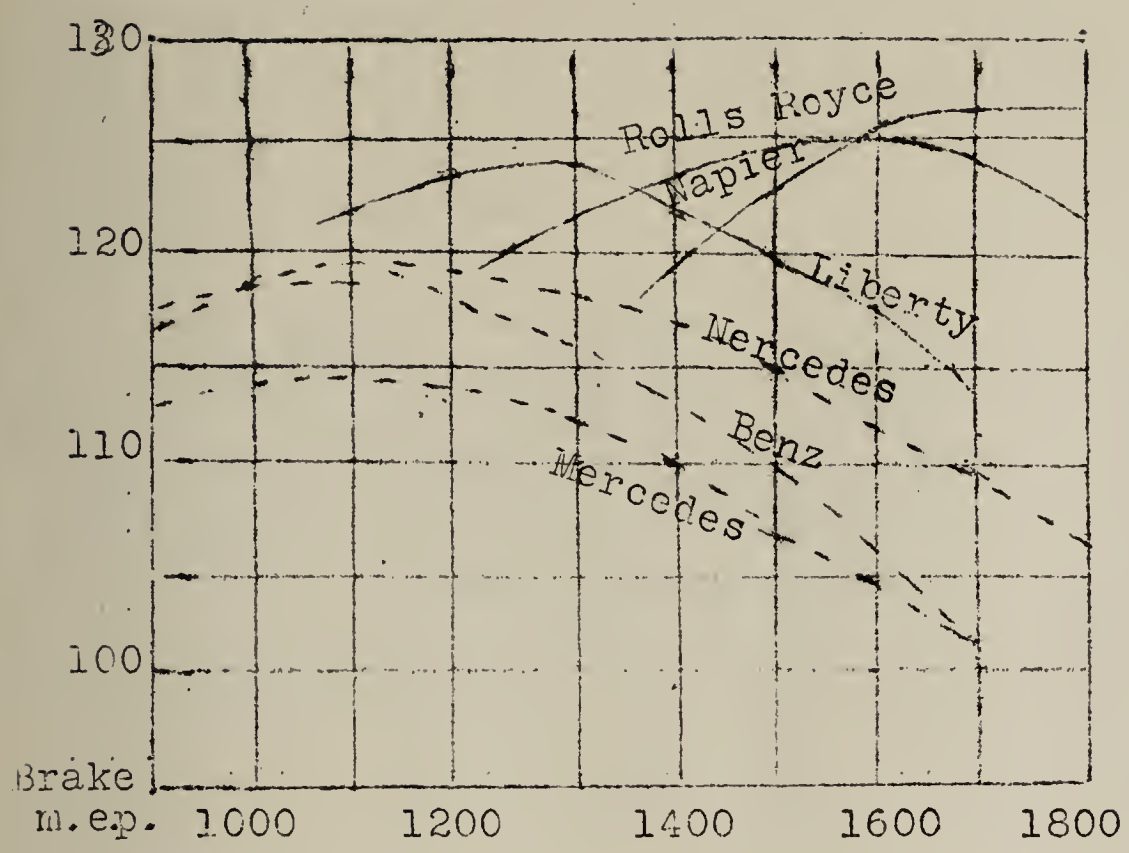
Curve A is Ricardo's Ideal Indicated m.e.p. , Curve B is the corresponding brake m.e.p. Curve C gives values for brake m.e.p. proposed by Berriman, and represents the relation,- brake m.e.p. = 20 X (compression ratio). The curve appears to have close relationship to results of actual tests.

Effect of Engine Speed. Effect of increase in engine speed, up to a certain point, is to increase the brake m.e.p., beyond which point the brake m.e.p. diminishes.

Effect of increasing the engine speed beyond a certain point is to reduce the mechanical efficiency because of higher engine losses, and to bring-in excessive stresses. For a particular engine, there is a limiting speed which is dependent on the factors of safety used in the design.

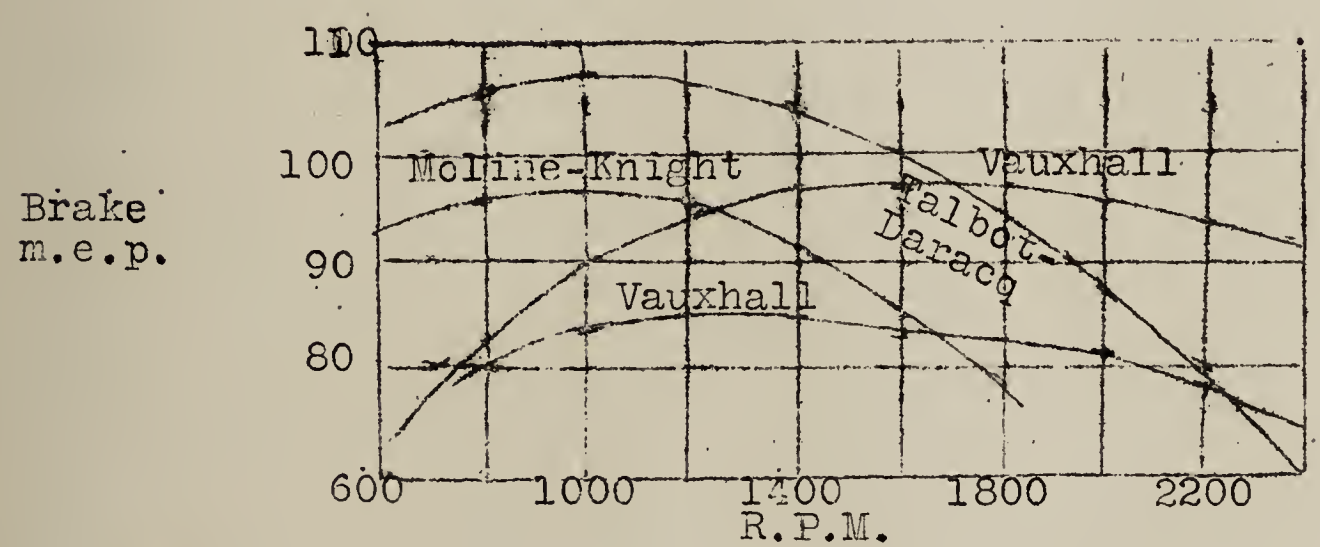
Reason for the increase in the brake m.e.p. with speed is because a greater amount of charge is admitted as the throttle is opened, and the relatively lower cooling losses at the more appropriate engine speeds. Beyond certain point, the effect of increasing the speed is to lower the volumetric efficiency, and also to reduce the mechanical efficiency at a greater rate than at lower speeds. Extremely high engine speeds are usually accompanied by higher working temperatures, so that the cooling losses are larger.

Curves are shown for water-cooled aircraft engines, in the design of which the maximum power output for a given cylinder capacity or weight was aimed-at.



Brake mean effective pressures are appreciably higher than those earlier attempted,- maximum about 114 to 126 lbs. per sq. in. Maximum values for the Benz, and Mercedes engines (6-cylinder, vertical-type) occur at about 1100 r.p.m. The Liberty gives its maximum brake m.ep. at about 1200 r.p.m.; the Rolls-Royce and Napier from 1600 to 1800 r.p.m.

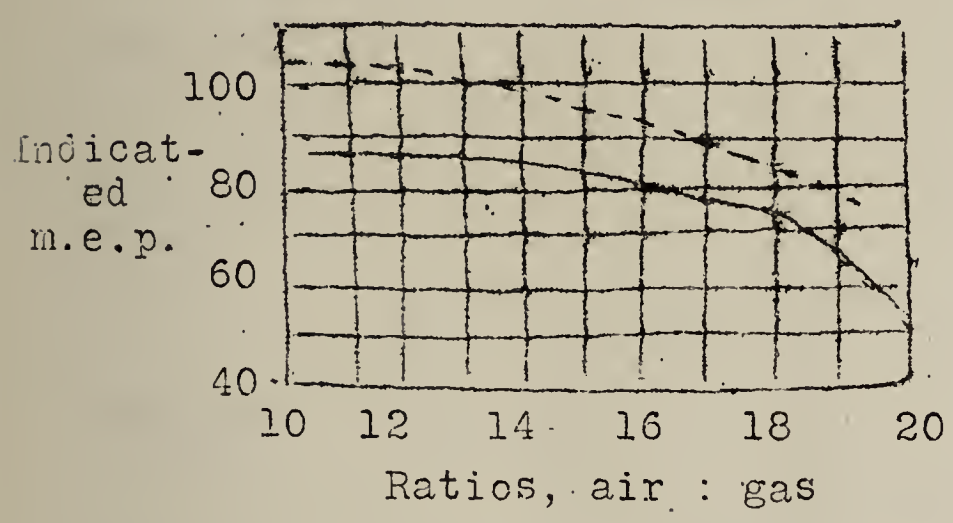
Curves are also shown for motor-car engines fairly typical of present practice.



Modern automobile engine progress appears to be following lines of development of the aircraft engines,- that is toward higher mean effective pressures and engine speeds.

Piston Speeds. With the aircraft engines, maximum values for brake m.e.p. are developed at piston speeds from 1200 to 1500 ft. per min. In modern motor-car engines the maximum brake m.e.p. is reached at piston speeds from 1200 to 1800 ft. per min. Gain in piston speed is largely due to improved design, materials and lubrication.

Effect of Mixture Strength. Manner in which mean effective pressure varies with mixture strengths is shown in the curve. Full line represents results of test by Dr. Watson on a four-cylinder motor-car engine, dotted line gives results of experiments on a sleeve-valve engine.



With weak mixtures, mean effective pressures are low; as the mixture is enriched, the m.e.p. rises and reaches a maximum for a mixture strength richer than that for perfect combustion (14.7). It is possible that richer mixtures supplied by carburetor are necessary to counteract the effects of the exhaust dilution. Mixtures which give the maximum power are rich mixtures.

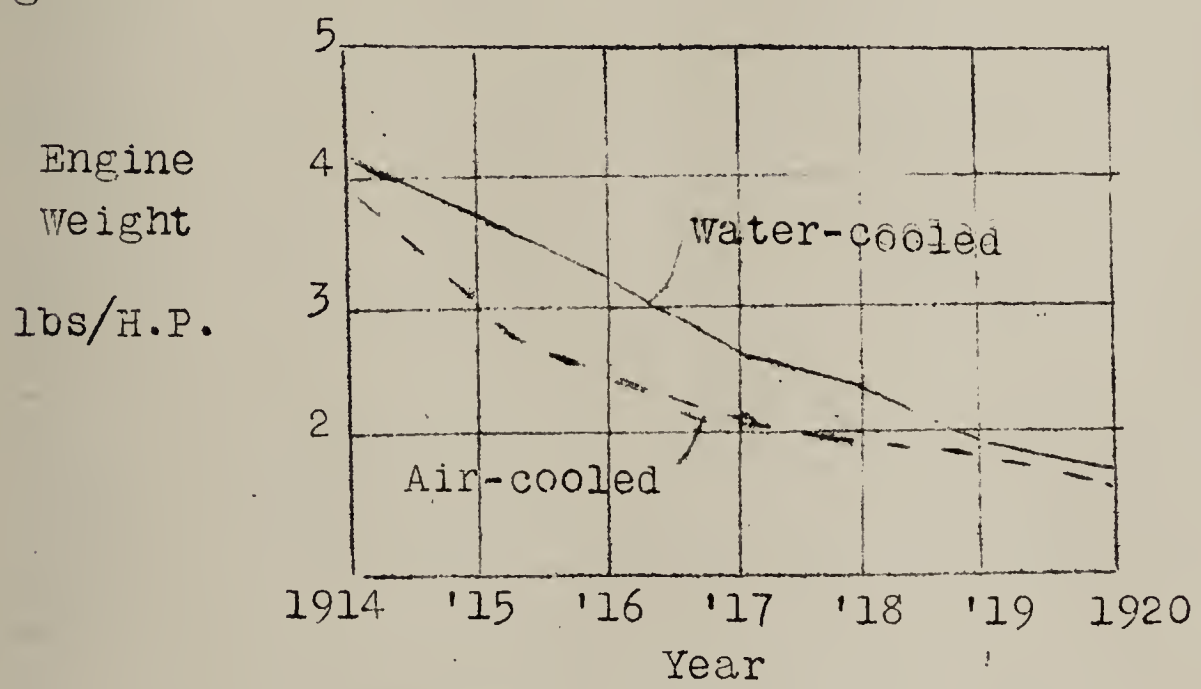
Weight of Engines

In airplane of light motor-car engine practice, minimum weight for a given power output is of vital importance.

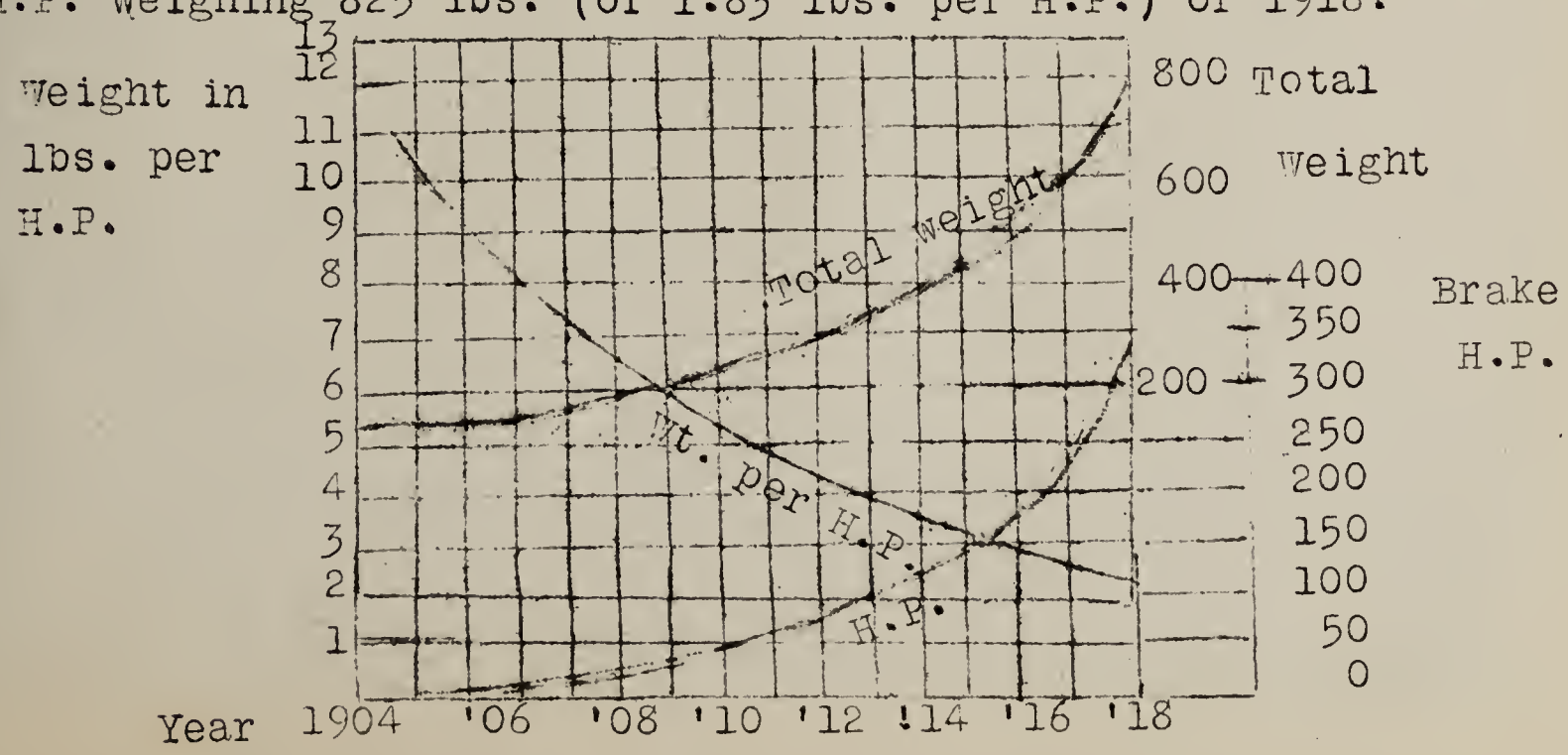
Weight per horse power of modern engines has been considerably reduced. Chief saving in weight is due to better design and the use of better materials.

Importance of minimum weight per H.P. in aircraft engines has resulted in the evolution of very light engines of all types.

Chart illustrating progress of aircraft engine-weight economy from 1914 to 1919 for both water-cooled and air-cooled engines.



A set of curves somewhat similar to the preceding chart, which was issued by the U.S. Advisory Committee for Aeronautics illustrates the development of aircraft engines from the time of the Wright Bros' 12-H.P. engine weighing 152 lbs. (or 12.7 lbs. per H.P.) in 1903 down to the Liberty 12-cylinder engine of 450 H.P. weighing 825 lbs. (or 1.83 lbs. per H.P.) of 1918.



Particulars and Performances of Typical Aircraft and Automobile Engines

No.	Type of Engine.	Cooling.	No. of Cylinders.	Diam. in Mm.	Stroke in Mm.	Compression Ratio	H.P.	Brake M.E.P.	R.P.M.	Power Speed.	Weight per B.H.P.	Weight per B.H.P. Per Hr.
								Lbs.		Lbs. per Min.		Lbs.
1	A. 100 Mono. Gnome (rotary)	Air	9	110	150	4·9	105/119	88/100	1,200	1,180	2·7	0·80
2	110 Le Rhone („)	„	9	112	170	4·8	114	87	1,200	1,310	2·8	0·60
3	110 Clerget („)	„	9									
4	B.R.1. („)	„	9	120	170	5·3	158	95	1,250	—	2·6	·63
5	B.R.2 („)	„	9	140	180	5·3	230	92	1,300	—	2·16	·63
6	B. 100 Anzani static radial	Air	10	105	145	4·5	95/100	86	1,200	1,140	4·2	·530
7	A.B.C. Gnat „ „	„	2	110	120	3·8	45	138	1,920	—	1·82	·520
8	A.B.C. Wasp „ „	„	7	115	150	4·1	170	112	1,800	—	1·53	·560
9	A.B.C. Dragonfly „	„	9	140	165	4·9	320	110	1,650	—	2·07	·585
10												
11												
12	C. R.A.F. 1A Vee type, 90°	Air	8	100	140	4·3	108	84	1,800	—	4·17	·603
13												
14	R.A.F. 1B „ 90°	„	8	105	140	4·24	110	82	1,800	—	4·09	·675
15	R.A.F. 4A „ 60°	„	12	100	140	5·3	160	89	1,800		4·17	·603
16	R.A.F. 3A „ 60°	Water	12	114·3	139·7	5·3	260	112	1,750	—	3·00	·500
17	Renault 75 „ 90°	Air	8	100	120	3·8	81	87	1,800	—	5·5	·650
18	190 Roll's-Royce „ 60°	Water	12	101·5	146·1	5·15	230	100	2,000		3·1	·530
19	250 „ „ 60°	„	12	114·3	165·1	4·53	265	95	1,800	1,730	3·4	·550
20	280 „ „ 60°	„	12	114·3	165·1	4·9	306	109	1,800	—	2·95	·530
21	300 „ „ 60°	„	12	114·3	165·1	4·9	320	114	1,800	—	2·95	·510
22	150 Sunbeam „ 90°	„	8	95	135	5·2	168	13[illegible]	2,100		3·42	·510
23	200 „ „ 90°	„	12	92	135	5·2	226	131·5	2,100	1,860	3·3	·510
24	250 „ „ 60°	„	12	100	135	5·3	270	128	2,100	1,970	3·3	·520
25	320 „ „ 60°	„	12	100	135	5·0	363	123	2,300	2,210	2·8	·510
26	430 „ Y or Arrow	„	18	110	160	5·0	—					
27	400 Napier Lion, „	„	12	139·7	130·2	5·55	450	122	2,000	—	840	1·85	·504
28	400 Liberty 12, Vee type, 45°	„	12	127·0	177·8	4·78 / 5·30	405	118	—	—	824	2·04	·504
28A	650 F.I.A.T., „	„	12	170	210	4·97	600	96·8	1,500	2,070	1,740	2·68	—
29	200 Hispano Suiza, Vee-type, 90°	Water	8	120	130	5·30	165	111	1,600		440	2·67	·540
30	150 „ „ „	„	8	120	130	4·70	220	121	2,000	—	515	2[illegible]	[illegible]
31	600 Lorraine Dietrich, Vee-type, 60	„	12	126	200	5·0	600	13[illegible]	1,[illegible]		1,200	2·10	
32	1000 „ „ 3-fan type	„	24	126	200	5·0	1,000	125	1,[illegible]	—	1,750	1[illegible]	
33	D. 160 Beardmore (vertical)	Water	6	142	176	4·55	180	10[illegible]	1,550		6[illegible]	4·45	[illegible]
34	230 Siddeley Puma (vertical)	„	6	145	190	4·95	250	1[illegible]	1,400		64[illegible]	2·58	[illegible]
35	300 Fiat (vertical)	„	6	160	180	4·7[illegible]	310	11[illegible]	1,600		52[illegible]	1·60	[illegible]
36	150 Benz („)	„	6	130	180	4·50	1[illegible]0	11[illegible]	1,200	1,420	5[illegible]	1·9	[illegible]
37	260 Mercedes („)	„	6	160	180	4·94	252	107·5	1,400	1,6[illegible]	[illegible]	1·7	[illegible]
38	180 „ („)	„	6	140	160	4·64	174	109·1	1,400	—	12[illegible]	1·6[illegible]	[illegible]
39	180 Maybach („)	„	6	160	170	4·60	180	95	1,200	1,240	—		

Aircraft Engine Weight.

The table (page 91) gives particulars of weights of modern aircraft engines in the "drystate", that is, the engine complete with magneto, and carburetors, but without water, fuel or oil.

If it is necessary to take the weight of the fuel into account, an allowance of 0.6 to 0.7 lbs. per H.P.-hr. should be made in the case of rotary air-cooled engines; 0.5 to 0.6 lbs. per H.P.-hr for static radial engines; and 0.5 to 0.6 for water-cooled engines.

Most of the larger sizes of water-cooled engines, from 300 to 600 B.H.P., show gasoline consumptions of from 0.49 to 0.51 lbs. per H.P.

The oil consumptions vary from 0.10 to 0.14 lbs. per H.P.-hr., in the case of rotary engines, down to 0.03 to 0.04 lbs. per H.P.-hr. for water-cooled engines, The larger water-cooled engines of 350 to 600 B.H.P., show oil consumptions of from 0.025 to 0.030 lbs. per H.P. hr. Static radial engines vary from 0.05 to 0.08 lbs. per H.P.-hr.

The weight allowance for radiators varies from about 0.15 to 0.40 lbs. per H.P., according to the class of machine the smaller value being appropriate for fast scout and two-seater machines, with efficient radiators in the slip-stream, and the larger value for slower machines of smaller horsepower. For machines of less than 100 h.p., the radiator weights vary from 0.4 to 0.7 lbs. per H.P.

The weight of cooling water carried in the cooling systems varies from 0.10 to 0.25 lbs. per H.P. in the case of fast scouts with maximum speeds fo 120 to 160 miles per hr., and from 0.40 to 0.50 lbs. per H.P. for slower machines (80 to 100 miles per hr.) of lower horsepowers.

A good approximate value for normal water-cooled engines of 250 to 450 B.H.P., for the combined radiator and water weights, is 0.6 lbs. per H.P.

Recent weight reduction in aeroplane engines has been small, - power units designed more for endurance and long life than for minimum weight alone.

Weight Economy and Engine Size. For engines of a given type, the larger sizes are the lightest per H.P. due to many factors, such as the multiplicity of cylinders, reduced crankshaft length, constant accessory weight, etc.

In the case of the Gnome series of engines, the 50 h.p. type weighed 3.4 lbs. per H.P., whereas the 160-h.p. type weighed 1.95 lbs. per H.P.

Again, in the case of the statis radial Anzani engines, the 25 h.p. type weighed 4.72, and the 200 h,p. type 3.4 lbs. per H.P.

Graphical method of illustrating reduction of weight per H.P.

Analysis of Aircraft Engine Weights. Component weights of several aircraft engines of recognized design are given, so that the distribution of the weight and the individual weight data may be examined in detail.

Weight data for a 230 h.p. six-cylinder vertical water-cooled engine of 145-m.m. bore and 190 m.m. stroke giving its rated H.P. at 1,400 r.p.m. is shown in the table.

Compression ratio 4.91, giving brake m.e.p. of 113 lbs. per sq. in.

Weight of dry engine, less exhaust pipes	850 lbs.
Weight per B.H.P.	3.68 "
Weight of water carried	30.9 "
Weight of radiators, empty	136 "
Weight of gasoline used per hour	135 "
Weight of oil used per hour	5.06 "
Gross weight of engine in running order, less fuel and oil	996 "
per B.H.P.	4.33 "
Gross engine weight, with fuel and oil for 6 hours	1,921 "
per B.H.P.	8.35 "

Component and Percentage Weights for 230 H.P. Aircraft Engine	Weight in lbs.	Percentage of Total Wt.
Cylinders, complete, including supports for rocker levers (6 @ 7.62 lbs. each)	265.5	30.753
Pistons complete with rings and wrist pins (6 @ 7.62 lbs. each)	45.72	5.296
Connecting rods complete (6 @ 7.13 lbs. each)	42.78	4.955
Valves and springs complete (24 @ 1.14 lbs.each)	27.36	3.169
Tappets, push rods, and rocker levers (12 sets @ 3.2 lbs. each)	38.4	4.447
Induction pipes complete (2 @ 2.625 lbs. each)	5.25	.608
Carburettors complete (2 @ 6.54 lbs. each)	13.08	1.515
Crank chamber, top half	103.25	11.959
Crank chamber bottom half (including cylr. studs and oil pump shaft)	110.25	12.77
Crankshaft complete	109.25	12.65
Propeller Boss complete	19.32	2.236
Camshafts, complete with bearings, less gear wheels (2 @ 8.25 lbs. each)	16.5	1.912
Magnetos complete (2 @ 10.75 lbs. each)	21.5	2.490
Ignition wiring complete (2 @ 2.12 lbs. each)	4.24	.491
Water pump complete	8.75	1.013
Oil pump complete	2.89	.335
Gasoline pump complete	6.75	.782
Exhaust manifold	15	1.737
Miscellaneous parts	7.53	.872
Total weight of engine	863.32	100

With another six cylinder engine of 180 h.p., water-cooled, the weight of base cylinders expressed as a percentage of the total was 17.5; that of the pistons complete with rings and wrist pins, 6.23; the connecting rods, complete with bushings, 4.55; the crankshaft, 10.60; the camshaft, 1.17; top-half of crank-case; 10.92; bottom half of crank case, 15.16; dual carburetor, 2.54; water-pump, 1.17; magneto (two), 4.25; and the twelve valves, complete with springs, etc., 4.55 percent.

Table 5.—Weights in Pounds of Engine Parts

	Liberty		Packard		Hispano-Suiza		Austro-Daimler	Bassé-Selve	Benz	Maybach	Mercedes
Number of cylinders	6	12	8	12	8	8	6	6	6	6	6
Rated horse power	200	400	180	280	180	300	200	270	200	300	180
Cylinders complete with valves	115.2	241.2	124.6	189.9	152.3	210.0	131.3	232.7	185.5	217.2	131.2
Crankshaft assembly including thrust bearing	104.4	105.8	64.8	86.5	53.7	73.4	101.8	138.5	87.0	117.6	70.0
Upper crankcase assembly with bearings, studs, etc.	77.2	92.0	60.3	80.7	33.8	52.6	133.4	} 209.5	85.6	94.3	} 172.3
Lower crankcase assembly	69.3	70.0	48.3	61.7	44.3	52.8	73.5		75.0	96.6	
Valve gear assembly including drive shafts, housings and accessory drives	44.6	90.2	73.8	95.5	26.7	31.9	74.8	82.3	39.0	74.2	70.4
Pistons complete with pins and rings	28.1	58.8	31.1	50.4	28.2	48.2	25.1	37.1	30.1	84.3	41.1
Connecting rod assembly including bearings:											
Forked	.....	34.2	17.6	26.2	16.0	22.9					
Plain	28.8	19.2	8.7	15.0	9.7	12.6	29.0	54.0	35.0	49.1	30.0
Propeller hub assembly	22.1	26.6	18.6	18.6	18.7	21.7	11.3	29.4	17.3	21.0	12.5
Ignition assembly:											
Generator	12.5	11.6	12.1	12.1							
Storage battery	10.5	10.5	10.5	10.5							
Distributor, wires, conduit	13.5	19.8	15.0	18.2	5.2	6.3					
Spark plugs	2.2	4.4	2.9	4.1	2.9	2.9					
Switch and voltage regulator	2.4	2.4	2.1	2.1							
Magnetos, drive gear and housing	.....	.....	.....	.....	43.1	40.7					
Total	41.1	48.7	42.6	47.0	51.2	49.9	35.6	33.1	29.3	26.2	32.6
Carburetor system:											
Carburetor	11.7	19.0	8.7	9.5	7.1	10.7	24.1	4.0	18.7	32.0	16.7
Intake header assembly	11.2	26.4	17.3	25.0	7.5	11.7	8.5	4.0	6.0	11.1	10.0
Water pump and connections	11.5	16.9	10.5	13.2	5.6	10.3	14.2	8.0	8.4	8.5	11.0
Oil pump and connections	9.0	7.5	5.4	5.4	3.6	10.5	16.4	9.6	5.7	13.0	13.2
Fuel pump	.....	.....	.....	.....	1.3	.....	1.6	.....	3.9	3.5	4.8
Miscellaneous	6.2	.....	8.2	8.3	.....	.....	28.1	27.8	11.7	7.0	31.8
Self starter	.....	.....	.....	.....	.....	.....	.....	.....	.....	6.0	
Total dry weight	580.0	856.0	542.0	933.0	476.0	632.0	708.5	870.0	638.2	871.7	647.0
Weight of water in engine	21.0	45.0	25.7	41.0	44.0	58.0	.....	.....	19.0		

The table on page 95 gives weights of engine parts of the principal American and foreign airplane engines.

The data in the two following tables show that for engines of the same horsepower the rotary and static radial air-cooled engines are the lightest.

For example, the 170 h.p. A B C "Wasp" weighs only 1.53 lbs. per H.P.; the 200 B.R.2 engine, 2.0 lbs. per H.P.; and the 450 h.p. "Cosmos" engine, only 1.47 lbs. per H.P.

The 160 Beardmore water-cooled engine weighs 3.3 lbs. per H.P.; the 160 Mercedes, 3.86 lbs. per H.P.; the 600 h.p. Fiat, 2.36 lbs. per H.P.; the 400 Liberty, 2.0 lbs. per H.P.; and the 350 Rolls-Royce, about 2.6 lbs. per H.P.

In the larger sizes above 300 h.p. there is not much difference between the air-cooled and water-cooled engine weights, although the air-cooled weights, on the average of present-day practice, show an advantage of about 5 to 8 per cent for engines up to 500 h.p.

The total weight of engine, oil, and fuel for a given number of hours' flight may be the best criterion, -man advantage in oil and fuel economy may in long flights quite outweigh an initial dry-engine weight excess.

The following table illustrates this point in air-cooled and water-cooled engines of about the same approximate horsepower.

Type of Engine	B.H.P.	Weight lbs.	Oil per hr lbs.	Gasoline per hr. lbs.	Total Wet-Weight in Lbs.			
					1 hrs flt.	3 hrs flt.	6 hrs flt.	10 hrs. flt.
Air-cooled static	450	680	24.5	270	1004	1651	2622	3920
450 Liberty, water-cooled	450	835	13.0	204	1326	1805	2522	3475
Air-cooled rotary B.R.2	230	500	20.5	155	694	1078	1657	2430
Rolls-Royce water-cooled	275	715	8.00	132	1054	1361	1822	2430

In connection with the last two engines in the table, it will be noticed that the engines are not the same horsepower so that, if the relative weights per horsepower are considered, the watercooled type is appreciably the better for the longer flights, as the following results show.

Engine	B.H.P.	Total "Wet" Weight in lbs. per B.H.P. for flight of			
		1 hour	3 hours	6 hours	10 hours
Air-cooled rotary, B.R.2	230	3.05	4.68	7.20	10.58
Rolls-Royce water-cooled	275	3.84	4.95	6.64	8.84

It is apparent that the water-cooled engine, with its better fuel and oil economy, and its greater endurance is the more suitable for long-distance flights.

Minimum Weight per Horsepower. To obtain minimum weight per horsepower, the engine must have minimum weight per cu.ft. of piston displacement per revolution, and must operate with maximum power per cu. ft. of cylinder volume.

This demands a combination of the maximum attainable mean effective pressure with high engine-speed. Limiting speed of falling-off of m.e.p. should be made as high as possible. Large valve openings and short inlet and exhaust pipes of ample size are needed.

The airplane engine must be multi-cylindered.

Weight of the engine per cylinder diminishes with increase of number of cylinders in line.

A saving in the weight of the crank shaft and crankcase is gained through the use of the V or W arrangement of cylinders; and a still further saving by the adoption of the radial arrangement.

Weight increases with increase of length of stroke.

Weight of the engine per cu.ft. of piston displacement per revolution is a function of the ratio of length of connecting-rod to stroke,- the shorter the connecting rod the less the overall dimensions and weight of the engine.. The objections to a short connecting rod have been pointed out.

The weight per cu. in. of piston displacement per stroke is a measure of the success of the designer in keeping-down weight.

MECHANICAL DESIGN

Careful theoretical study of both the mechanical and thermodynamic problems involved, and intensive research have resulted in the production of light high-speed engines which, besides giving an almost incredible power output for their size and weight, show as high efficiencies as the largest slow-speed types. The basic principles, both mechanical and thermodynamic, upon which the performance of the engine depends, have been investigated in so complete and comprehensive a manner that the performance of an engine can be predicted with considerable accuracy from a study of the design; or an engine can be designed to fulfill any specific requirement as to power output or efficiency with considerable precision.

In the design of an airplane engine the aesthetic side must not be overlooked. Beauty of form and of proportion is a guide to mechanical correctness; the appeal of a design to the aesthetic sense is frequently as reliable a guide as is a mechanical analysis of its mechanical features.

Prevalent but quite erroneous belief that the reliability and the efficiency of an engine are to a great extent a function of the actual number of parts that the engine contains,- a great fallacy. While the number of parts should be kept down to a minimum compatible with efficiency and mechanical correctness, the reduction of number of parts can be easily overdone. Fewness of parts too often denotes excess of compromise. All design must be based on compromise, and it is upon the soundness of judgment by which the compromise is arrived-at that the success of the engine ultimately depends.

Illustrations: Use of universal joints; and duplication of exhaust valves.

Designer has to decide whether he will risk a simple expedient or resort to a more complicated construction. It is better to use twenty parts if needed to comply with the laws of sound mechanics than to defy these laws by the use of a single part. The aero-engine which, during the war, was capable of running for the longest period without overhaul, contained at least 50% more parts than any other engine in the service. That increased number of parts necessarily involves increased maintenance is a fallacy. Fewness

of parts may reduce manufacturing costs to a small extent; the amount of fitting work may be universely proportional to the number of parts, and fitting work is very expensive. With well-thought-out design and accurate machine work, it should be possible to eliminate hand-fitting almost entirely. Hand-fitting is most effective barrier to interchangeability.

Highly specialized materials are not generally necessary, except in especially closely limited cases.

Extreme rigidity is a prime necessity in high-speed engine design.

Classification of Engine Parts

(1) The Power System
(2) The Fuel and Carbureting System
(3) The Ignition System
(4) The Lubrication System
(5) The Cooling System.

Successful operation of an engine depends on these systems working well together.

Power System. The harnessing of the power generated and delivering it in useful form; function of engine parts must be performed exactly on time, valve design and timing important. One half the number of cylinders represents the number of cycles completed and the number of power impulses per revolution. More even flow of power as number of cylinders is increased. Rapddity with which events of cycle take-place.

Illustrations of distribution of power.

Engine must withstand severe shocks; parts work under high temperatures,- importance of correct design, choice of materials, treatment of materials, workmanship.

The general design of the engine and the various parts will be taken-up later.

The Fuel and Carbureting System. Fuel supply with its controls forms vital part of the machine; must be designed to meet requirements of the particular fuel used.

Supply tanks, piping, pumps, valves, strainers, joints and fittings require proper consideration to ensure good operation.

Requirements of the fuel system are given in "Handbook of Instructions for Airplane Designers" prepared by the Engineering Division of the U. S. Air Service.

Carburetion taken up next term.

The Ignition System (Taken next term).

The Lubrication System. Internal-combustion engine most difficult to lubricate because of exposure of lubricated surfaces to high temperatures; lubrication has to be maintained under severe conditions.

Oils. Choice of best oil difficult. To maintain proper film between the surface the oil must be of correct body and character to meet the lubricating requirements, and of correct quality to withstand the high temperature. Full power is not secured unless a piston seal is maintained. Many engine troubles due to faulty lubrication; worn bearings, scored cylinders, carbon deposit, excessive quantity.

If oil is correct for the engine, there will result,- less frequent repairs, full power, saving of gasoline and oil, small carbon deposit. Oil problem most difficult for exact solution.

Physical tests give an index to certain properties,- "viscosity" test gives some idea of the body and tendency to resist flowing; "flash and fire" test give some idea of suitability for known temperature; "cold" or "pour" test shows lowest temperature of flowing. These tests do not determine the efficiency of an oil. Best solution of problem through practical experience.

Lubrication requirements of different engines differ widely. The following factors should be considered: type of engine; size; valve construction; piston rings and fit; piston design and material, clearance; cooling system, air or water; speed; and climatic conditions. No one oil will correctly lubricate all engines.

Correct lubricant for a given engine should be an oil,- of highest lubricating efficiency, whose body and viscosity are

best suited, of sufficient body and heat-test to retain the oil film, of sufficient body to resist squeezing-out.

Reasons for using oils of greater fluidity in cold weather,- greater ease in starting; suitable for certain types of pump and pump locations and exposed piping

When oils of great fluidity are not necessary, oil of less fluidity should be used,- former is limited in lubricating qualities.

Draining-off used oil and replacing fresh oil is an important detail in efficient operation. Oil becomes thinned-down in use, and its lubricating efficiency is impaired. Usually changed after five to twenty hours of flight. Used oil can be reclaimed. Increased trouble in recent years due to character of gasoline; probability of gasoline in liquid form being drawn into cylinder. Amount of gasoline which mixes with this lubricating oil depends somewhat on the correctness of the oil.

When gasoline mixes with the oil there results:

An inefficient piston seal with consequent high consumption of gasoline and oil, and loss of power;
Premature wear of cylinders, pistons, crank-shaft bearings, and other parts;
Excessive carbon deposit, causing pre-ignition;
Burned-out bearings;
Low compression;
Excessive accumulation of carbon sediment.

To overcome such condition it is important

That the carburetor choke-valve be used sparingly.
That during cold weather a means be provided to enable the engine to operate at a higher temperature,
That the used oil be frequently drained-off and replenished with fuel oil;
That the correct oil be used.

Carbonization. Rapid carbonization results from use of poorly refined oils of inferior quality or oil of incorrect body and volitility; and from piston leakage. Rapidity often lessened through carburetor adjustment. Commonly called carbon deposits are not all carbon. Carbon deposits decrease compression space; promote pre-ignition.
Carbon deposit may be due to quality of gasoline.
Carbon removal.

Oil Used. Mineral oils are most generally used on account of their chemical stability which renders them less prone to carbonization or gumming. They do not, however, possess the property of "oilness" to same extent as vegetable or animal oils.

Chief virtue of animal and vegetable oils lies in their high "oilness" which of use in special cases, one being that of the rotating engine.

Their defect lies in their instability which renders them liable to become gummy and acid and which also causes them to carbonize more rapidly than mineral oils.

Blended oils containing a small percentage of oil other than mineral are fairly widely used; they seem to be desirable where conditions are somewhat severe.

Other factors being equal, the oil which gives a minimum rise of temperature to the bearings is the best to use. An oil of low viscosity causes greater friction because it squeezes out easier; an oil of higher viscosity results in increased fluid friction. The friction diminishes as the thickness of film increases.

One of the most troublesome features of lubrication is the decrease of viscosity with rise in temperature.

Standard specifications for lubricating oils for airplane engines have been adopted by the U. S. Army and Navy.

Table on page 104 gives the properties of representative American Lubricating Oils for use in internal-combustion engines,

PROPERTIES OF REPRESENTATIVE [illegible]
FOR USE IN INTERNAL COMBUSTION ENGINES

Kind of oil	Physical properties							
	Baumé grav.	Flash, deg. F.		Deg. F burn	Deg. F chill	[illegible]		
		Open cup	Closed cup					
Havoline:								
Light	25.9	370	380	430	33	[illegible]	[illegible]	[illegible]
Medium	25.0	385	395	450	34	247	[illegible]	[illegible]
Heavy	25.6	395	410	455	46	355	[illegible]	[illegible]
Mobiloil:								
"E" Light	28.1	370	380	420	0	167	[illegible]	[illegible]
"A" Medium	21.8	360	360	420	24	330	97	[illegible]
"B" Heavy	26.3	500	470	580	41	1,640	397	[illegible]
Arctic Lt. Med.	23.3	370	380	425	6	221	74	[illegible]
Arctic Medium	21.1	370	585	430	8	300	87	[illegible]
"BB" Med. Heavy	25.8	460	...	540	46	926	243	[illegible]
Monogram:								
Light	27.6	360	360	410	20	140.	60	41
Medium	26.0	375	370	430	23	289	95	50
Heavy	28.9	430	445	505	34	340	108	55
Ex. Heavy	24.7	465	425	535	58	1,583	356	110
Perfection:								
"A" Light	29.1	400	410	470	26	181	71	45
"B" Medium	24.9	390	400	450	32	243	81	47
"C" Heavy	29.3	420	430	495	40	316	103	54
Socony:								
Zero	24.3	395	410	470	81 at 0	219	77	46
Polarine Heavy	25.4	385	380	450	35	300	103	51
Texaco:								
Light	21.3	335	340	380	81 at 0	205	69	[illegible]
Medium	20.9	350	350	400	81 at 0	301	85	[illegible]
Heavy	19.3	356	360	420	10	495	[illegible]	[illegible]
Veedol:								
Aero No. 1	26.2	455	450	535	39	795	212	[illegible]
Aero No. 2	27.1	450	445	530	38	814	[illegible]	[illegible]
Aero No. 3	26.3	435	435	520	38	[illegible]	[illegible]	[illegible]
Aero No. 4	27.6	440	430	515	24	516	[illegible]	[illegible]
Aero No. 5	24.7	440	450	520	28	[illegible]	[illegible]	[illegible]
Aero No. 6	24.7	460	460	540	27	[illegible]	[illegible]	[illegible]
Zero Heavy	26.2	440	...	480	23	[illegible]	[illegible]	[illegible]
Zero Extra Heavy	26.1	460	...	530	44	[illegible]	[illegible]	[illegible]
Wolf's Head:								
Heavy	[illegible]	[illegible]	[illegible]	[illegible]	32	[illegible]	[illegible]	[illegible]
No. 8	[illegible]	[illegible]	[illegible]	[illegible]	46	[illegible]	[illegible]	[illegible]
Castor oil	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

The Cooling System

Heat which has to be removed from the cylinder in order to keep its temperature within the limit which permits satisfactory operation is about equalnto the heat equivalent of the work done in the cylinder, or about 40 B.t.u. per brake horse power per minute. This quantity will increase or decrease with change in operating conditions; the limits are between 30 and 60 B.t.u. per brake horse power per minute.

Air Cooling System. If large amount of radiating surface can be created by use of fins or projecting ribs, continuous running of the engine is secured provided a sufficient supply of cooling air reaches these extended surfaces. An induced flow of air created by mechanical means is sometimes needed Air cooling without artificial means of induced flow is common practice with rotating engines. Used almost exclusively in rotating and radial engines. In vertical or Vee engines with several cylinders in line, the cooling problem has been met by use of mechanical means of directing the air to the different cylinders. With the rotating and radial types the motion of the plane and the engine location in the slip stream ensure an adequate flow of air for cooling.

Water Cooling System. Water cooling is the general method employed on static engines. System made possible by circulating cooling water through the radiator which removes the greater portion of heat given to the water by the cylinders. Engine should not operate under an unfavorable temperature. There should be a correct balance between the radiator capacity and the power of the engine; the cooling system is a vital factor in the efficiency of the engine.

Thermosyphon Action. When engine is hot, some circulation is caused by thermosyphon action; force which promotes this circulation is small.

To assist such circulation, a pump is inserted in the pipe connection between the radiator and the inlet to the jackets.

With the pump circulation, circulation is positive and is established immediately on starting the engine.

For the purpose of increasing engine efficiency, thermostats have been fitted to the water circulating systems of some engines, which block the circulation of the water to the radiator and by-pass it until the water jackets of the cylinders have attained a predetermined temperature.

Jacket water is usually kept at as high a temperature as the boiling point will permit.

Water is not the ideal cooling agent; a less volatile fluid would permit higher cylinder temperature. Same result might be obtained by closing the system and operating under pressure but this necessitates heavier material for the radiator and increase in weight.

Air Cooling versus Water Cooling. The water-cooled engine has heavy handicap of a radiator and water connections, involving additional weight, and greater vulnerability. Against these defects must be offset a large advantage on score of reliability, and the ability to consume less oil, to employ a higher compression, and therefore to obtain a lower fuel consumption. The radial engine, by reason of the disposition of the cylinders and its relation to the slip stream offers ideal case for air-cooling.

Water cooling offers two advantages: (1) the transmission of heat from the cylinder is more rapid to water than to air, and (2) the ultimate cooling surface in the radiator can be made much greater than is possible at the cylinder. Air cooling has the advantage of reduced total weight and diminished vulnerability.

By water-cooling, it is possible to run with higher speeds and compression ratios than are practicable with air-cooled cylinders. The possible increase in speed and ratio of compression are relatively less important when compared with performance of best recent constructions in air-cooled cylinders.

With water cooling it is possible to maintain almost any desired cylinder temperature. The great virtue in water-cooling appears to lie not so much in the fact that the mean temperature of the cylinder can be kept fairly low, but rather in the fact that

water boils at a comparatively low temperature, and has a verybhigh latent heat, so that in the event of local overheating due to any cause, the water will boil locally, and the heat abstraction will be intensified to an enormous extent. In the air-cooled engine there is no such advantage; there is nothing but the conductivity of the metal to ensure against the development of hot spots. At best, the conductivity of any metal is but a poor substitute for the latent heat of evaporation of water. While the air-cooled cylinder may be kept cool enough to run satisfactorily as long as everything is in perfect condition, it has not the factor of safety of the water-cooled cylinder. For military purposes, and more especially for fighting machines, the advantages on the score of reduced vulnerability, freedom from freezing, and reduced weight, are all very important and may outweigh the disadvantages of air-cooling, but it is doubtful if air-cooling will replace water-cooling for long-distance military flight.

The resistance or drag of air-cooled cylinders is considerable but has not been determined satisfactorily. In the rotating engine there is, in addition to the drag, the resistance due to churning which reduces directly the b.h.p. of the engine. This resistance increases so rapidly with speed that it is not found desirable to operate rotating engines at speeds in excess of about 1,400 r.p.m.; the increase in indicated power which results from increased speed is largely used-up in over-coming the increased air resistance. The total work done by an air-cooled engine in overcoming air resistance is probably greater than the total work done by a water-cooled engine in overcoming the drag of its radiator. Until quite recently air-cooled cylinders were at a great disadvantage both in fuel economy and in power developed per unit volume of piston displacement, but recent constructions have put the air- and water-cooled engines nearly on a par in these respects.

Valuable research work on air-cooling by Major Norman and Professor Gibson at the Royal Aircraft Establishment, and fine results were obtained with experimental cylinders, having thin but well-ribbed steel barrels with very thick aluminium cylinder heads

cast-on, and with steel valve seats cast-in.

By the use of very thick aluminum the heat can be spread fairly uniformly over the cylinder-head and out to the deep and thick flanges, with the result that local over-heating can be avoided and results obtained comparable with those of a water-cooled cylinder.

It has, however, been found very difficult to ensure a sound job when casting the head on to a steel barrel, while the alternatives of bolting, screwing; or shrinking have none of them proved sufficiently reliable. Given that, as in a radial, or rotating engine, the position of the cylinder is such that it has the best of facilities for cooling, it has been demonstrated that the air-cooled cylinder will give from 90 to 95 per cent as high power and efficiency as a corresponding water-cooled cylinder, so long as there is no distortion, leakage, or detonation; but while the water-cooled cylinder can generally survive such ailments due to intensification of heat transference from the seat of the trouble, the air-cooled cylinder has no such advantage, and must give way to local overheating followed by preignition, and perhaps by distortion and seizure of the piston or burning-out of the valves.

The lack of recuperative power which is the weakness of the air-cooled engine applies also, though to a lesser extent, to those water-cooled engines in which there is a double metal wall through which the heat has to pass before reaching the cooling water.

Selection of Water. Soft water should be used in the cooling system. Hard waters contain lime and similar sediments which incrust the metal surfaces and prevent the heat from coming through to the water.

In cold weather it is best to drain the system when not in use, as all anti-freezing solutions are objectionable. Alcohol seems to be the most satisfactory; and a smaller percentage can be used with safety than that commonly given in the charts and tables.

Engine Lubrication

Most important parts of engine to be lubricated,- cylinder walls, crankshaft, crank-pin, piston-pin, and cam-shaft bearings.

Maintenance of adequate oil film necessitates efficient provision for removal of heat generated and for continuous replenishment of oil. Heat must be carried away by an excess of oil.

Pressure Oiling System. Splash systems are not satisfactory; wet sump is undesirable. Modern airplane engine has pressure oiling system.

Lubrication system must supply needs of lubrication and make use of oil as cooling agent. Advantages of forced lubrication mainly due to cooling effect of excess lubricant.

Objects sought are:

To circulate as much oil through the bearings as possible.
To keep the oil as cool as possible.

Amount of oil that can be circulated depends on pressure at which it is supplied, the clearance of bearings, and viscosity of the oil. Pressure is merely a measure of rate at which the oil is being circulated.

For free circulation with pressure feed lubrication, a minimum clearance of about 0.0015 in. should be given to crank shaft and connecting-rod bearings. Oil of low viscosity generally used.

Large flow of oil is needed. Capacity of pump should deliver at least 1/2 gallon of oil per hour for every brake horsepower. Amount of oil circulated in Liberty engine is about 12 gallons per minute, with temperature rise of about 10° F.

Oil consumption is amount of oil burned or otherwise lost,- varies from 1/50 to 1/20 lbs. per horsepower hour for static water-cooled engines, may be as much 3/20 lbs. for rotating engines.

Applications of the System. Description of a normal lubricating system. The Liberty engine, Hispano-Suiza engine, Curtiss K-12, Curtiss OX engine, Hall-Scott L-6 engine, Napier "Lion" engine, Fiat-650 engine, Benz-230 engine and Maybach engine.

Oil Pumps. Three types in general use: the gear pump, the plunger pump, and the valveless oscillating plunger pump. Typical examples on page 111.

The gear pump has the advantage of a purely rotary motion. It has disadvantages:

(1) Its volumetric efficiency falls very rapidly with wear.
(2) When slightly worn it has little or no suction lift.
(3) It is easily jammed by the entry of particles of foreign matter.
(4) It delivers a very rapidly pulsating flow and may liable to set-up vibrations in the oil piping.

The plunger pump has the advantage that its volumetric efficiency varies little with wear, and also that it has a high suction lift; but, on the other hand, the suction valve is liable to stick, and put the pump out of operation, it also limits the speed at which the pump will operate.

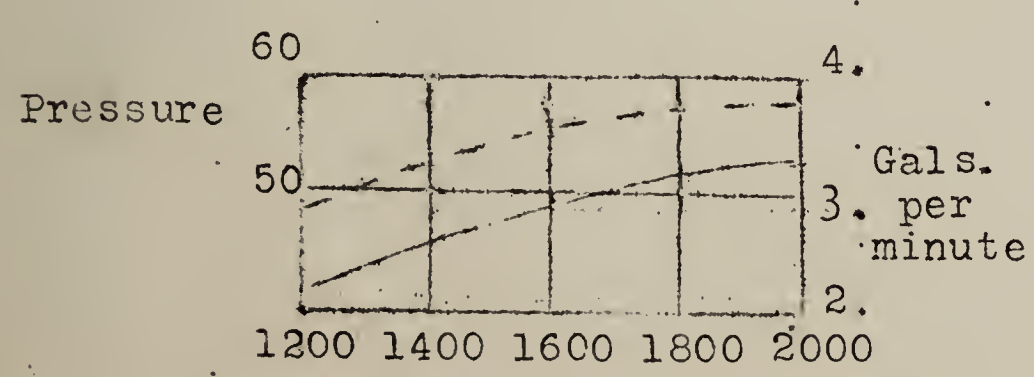
The valveless oscillating plunger pump is a satisfactory type,- no valves to stick or to limit speed of operation; also capable of dealing with dirt, particles of felt, etc. When run at high speeds it is necessary to provide an air chamber on the suction side.

Cons truction Details. Gear pump consists of a driving spur-gear which meshes closely with an exactly similar driven gear, The pump is driven in such directions that the incoming oil is carried around the gears and not between them. The pumping capacity of each gear is approximately equal to the space between half the number of teeth. Capacity can be increased by increasing the pitch.

Two scavenger gear pumps are sometimes combined in the form of three-gear pump,- a driven gear on each side of the central driving gear.

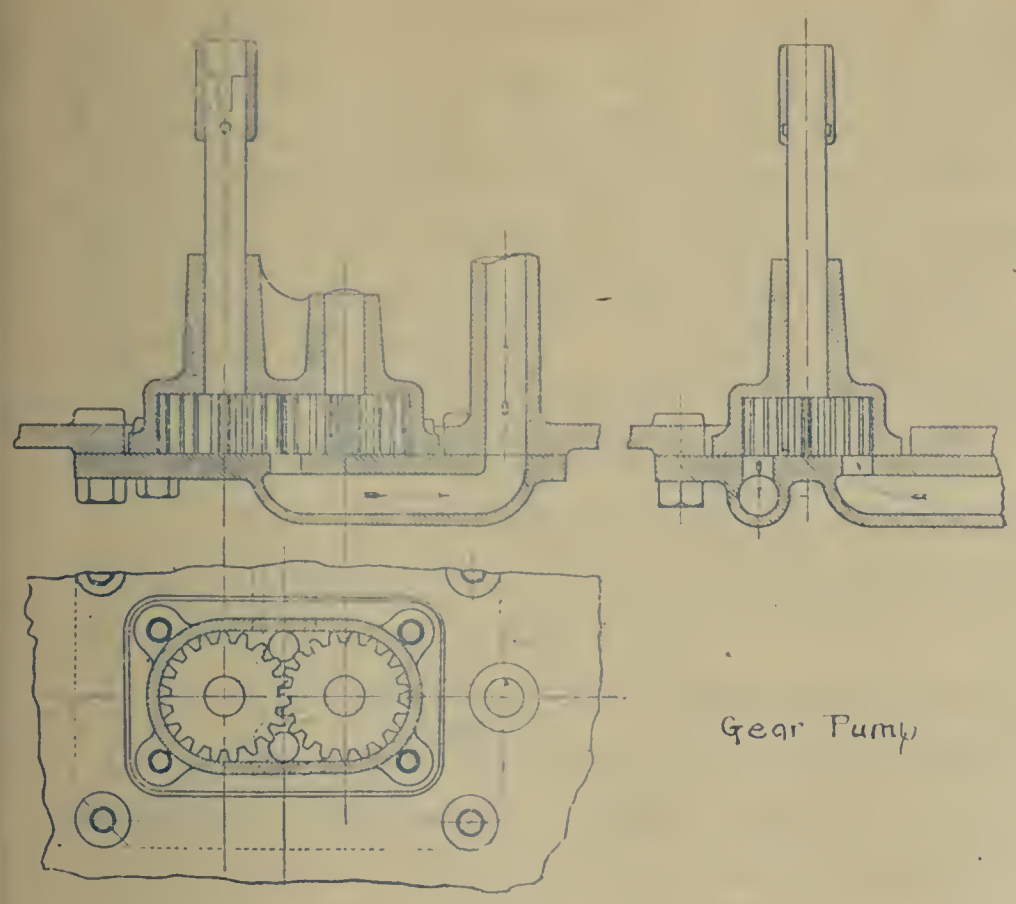
Scavenger and pressure pumps are sometimes combined in parallel planes by placing all the driving gears on the same shaft.

Curve showing performance of pressure gear-pump of the Packard-180.



Plunger pumps are sometimes combined by driving two plungers simultaneously from the same crank. Oscillating pumps may be similarly combined by having the plunger bodies in a single casting which is pivoted.

An eccentric sliding vane pump has been used to limited exten



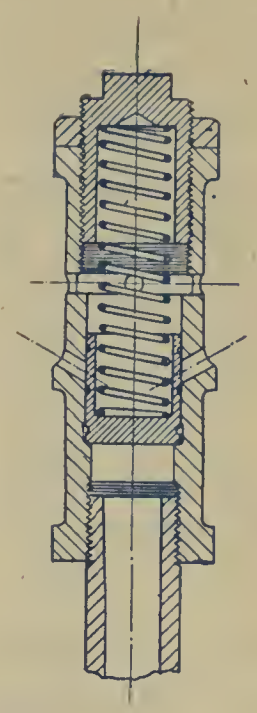
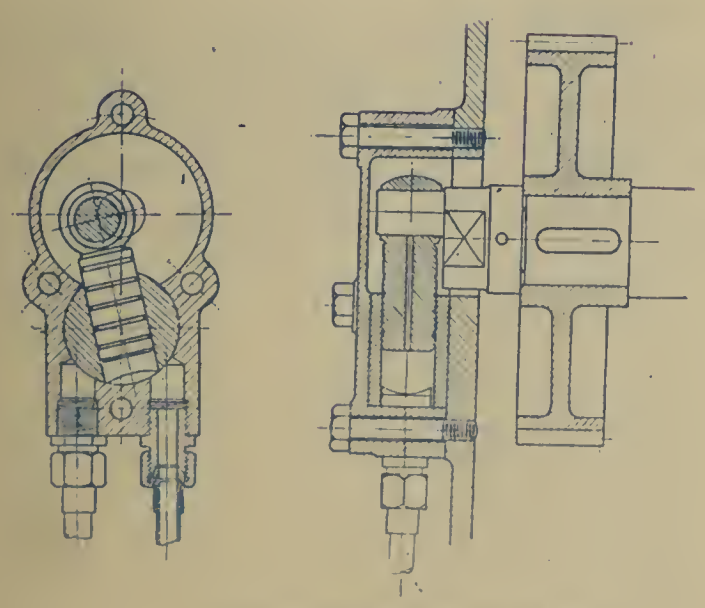
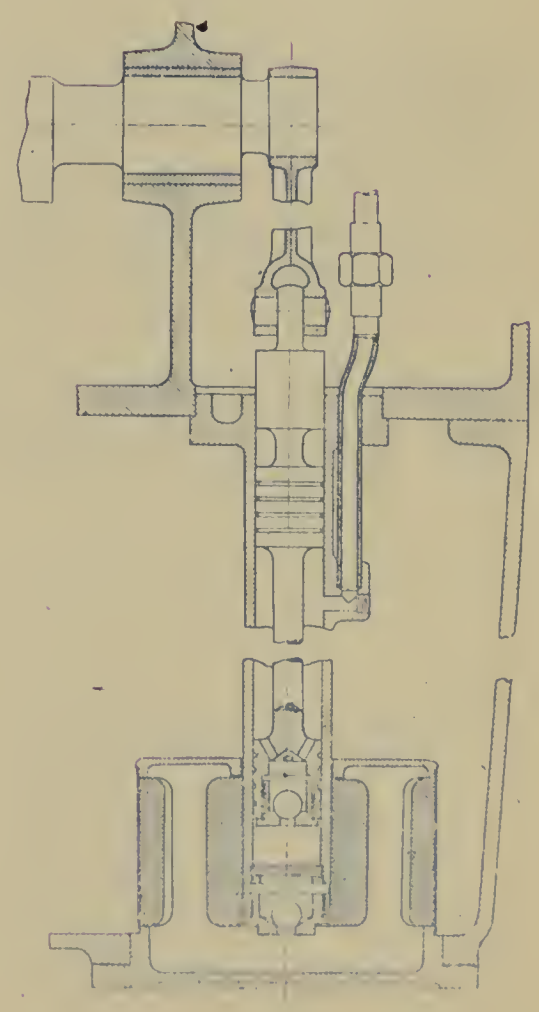


Fig. 63.—Plunger Type Oil Relief Valve





Oil Relief Valves. Pressure feed systems are provided with a relief valve, commonly set to relieve from 35-50 lbs. per sq.inch. When gear pumps are used and the flow of oil is practically continuous, almost any type of spring-loaded relief valve will serve, but when single-acting plunger pumps are used and the flow is pulsating, the design of the relief valve requires careful consideration. A form of relief valve which is said to give satisfactory results is shown. It has a swept volume equal to two or three times that of the pump. It is loaded by a long spring chamber for which steadies the pressure.

The relief valve should be fitted as remote from the pump as possible and the pressure gauge connection should be taken as near the relief as possible,

Oil Filters. An objection to pressure feed lubrication is that the circulation of a large quantity of oil involves the circulation also of a large quantity of grit which tends to rapid wear. The size of grit which causes wear cannot be filtered out with gauze. Desirable to provide a mesh filter capable of arresting large particles which might choke up the system, and which will not readily become choked itself.

Whether filter should be fitted on the suction or delivery side of the pump is an open question. If on the suction side it is liable to stop the flow of oil to the pump; if on the delivery side it is liable to burst. Best compromise appears to be to fit a coarse mesh on the suction side, and also a somewhat finer mesh on the pressure side.

Experiments have been carried out with centrifugal filters or separators which may prove capable of eliminating some of the finer particles of abrasive matter.

Valve-Gears

Valves mechanically operated by means of cams acting on them either directly or indirectly.

Camshafts. Camshafts may be placed near base of the cylinders and operate the valves through push-rods and rocker arms, or overhead camshafts may be used with the cams acting on the valves directly or through rocker arms. Illustrative examples.

In recent years the tendency has been to do away with push-rods and to use overhead camshafts. The arrangement reduces the weight and complexity of the valve gear; and, makes for better maintenance of the valve timing.

With overhead camshafts there may be either; (1) one camshaft over each row of cylinders with the cams acting directly on the valves, or (2) one camshaft with cams acting directly or one set of valves, and indirectly through rocker arms on the other set of valves, or (3) one camshaft with cams acting indirectly through rocker arms on both sets of valves, or (4) two camshafts with the cams acting directly on the two sets of valves. Illustrative examples.

Integral construction most generally used. Camshaft must be stiff and must be well supported, must not spring under shocks received; flexure should not exceed 0.002".

Number of bearings varies, and their lengths depend on arrangement. Front bearing which usually carries added load due to timing gears or chain is given greater length than other bearings.

Ball bearings have been used, but practice is not generally followed.

Width of cam not less than 1/4 diameter of valve for roller and not less than 1/3 diameter of valve for sliding follower.

Cams

Shape of cam depends on desired valve action and on form and location of follower,- flat tangential flanks; convex circular flanks, flanks changing from concave to convex.

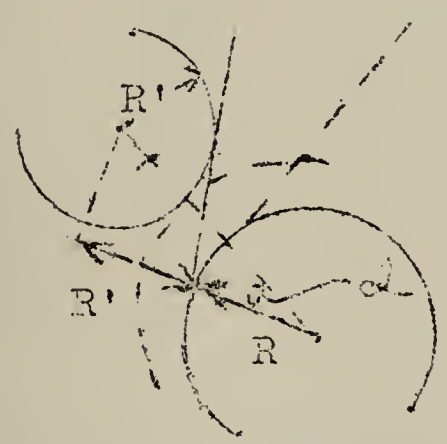
Cam followers,- roller, flat surface or rounded surface; may be fixed to valve plunger or may be mounted on a radius arm. Flat follower necessitates a convex flank for the cam.

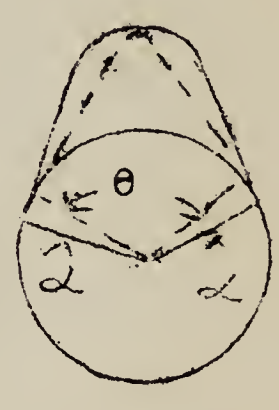
Common method of lay-out.

Correction for clearance in valve adjustment.

Tangential Cam.

$$\cos\alpha = \frac{R + R'}{R + (R' + cl)}$$



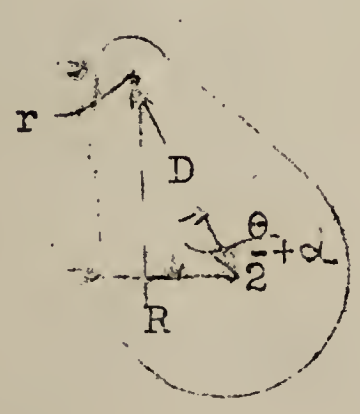


Symmetrical cam gives equal periods of time for opening and closing.

If line of motion of follower is off-set a symmetrical cam gives the period of time for closing less than the period of time for opening. For equal periods of time for each, the cam would be unsymmetrical,- its portion for closing will be greater than the portion for opening.

The valve is accelerated as long as the roller remains on the straight portion of the cam, and deceleration is during the period of time that the roller is in contact with the circular portion.

Definite relation between radius of base circle, R, radius of rounding circle, r, and the lift, l.

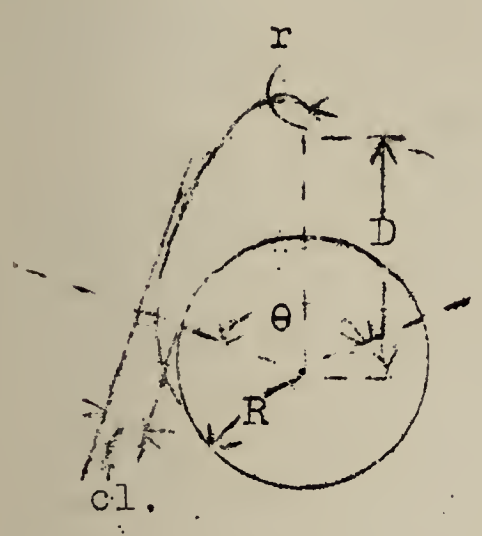


$$R = r + (l + cl)\frac{\cos(\frac{\theta}{2} + \alpha)}{1 - \cos(\frac{\theta}{2} + \alpha)}$$

$$r = R - (l + cl.)\frac{\cos(\frac{\theta}{2} + \alpha)}{1 - \cos(\frac{\theta}{2} + \alpha)}$$

If rounding circle is kept the same for exhaust cam, acceleration and decleration will be the same as for inlet valve, and there will be a period if dwell for full open position.

Rounding circle may be made of larger radius which will shorten period of acceleration and extend period of deceleration.



Mushroom Cam

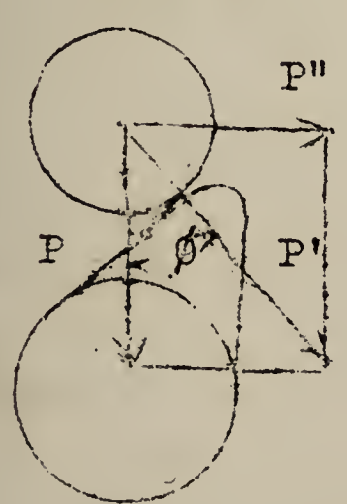
With mushroom follower there is no period of dwell. Relations for R and r may be same as for tangential cam. Flank is an arc of circle tangent to two circles, R and r, and to face of follower when raised the amount of the clearance.

Constant Acceleration Cam

Periods of acceleration and deceleration are independently variable; the two together may be equal to or less than 1/2 the period the valve is open. There may be period of dwell.

Comparison Curves Valve-lift curves do not permit true comparison of the different types of cams, because of certain assumed values. Variables are different in character; inpossible to give them values strictly equivalent for different types.

Side Thrust. Vertical pressure, P, is equal to sum of spring pressure and inertia force. Pressure at surface of cam, P', is vertical only at full lift and no lift. Side thrust on cam follower, P" is equal to



$$P'' = P \tan \phi$$

$$P' = \frac{P}{\cos \phi}$$

Side thrust may be troublesome

Normal pressure (P') and side thrust (P") increase more rapidly than radial force, (P), the quicker the opening of the valve.

Work Done by Cam. Cam must do work in compressing spring; work must be done in giving acceleration to valve and moving parts. Force of acceleration determined by the design of cam and follower and the mass accelerated. Example.

Cam Design.

Primary considerations are to open and close the valves as rapidly as possible with minimum of stress or noise, and to arrive at a form which can be readily produced.

Desirable to avoid concave surfaces for manufacturing reasons; by suitable combination of cam and follower, necessity for concave surfaces can be avoided.

For about first half of lift and latter half of closing, movement of valve is controlled by cam; from about half lift to full lift to about half closed, vavle is under control of spring. Acceleration permissible while valve is under spring control is governed by pressure and rating of spring; during first and last portions of valve movement, acceleration is governed by permissible pressure on flank of cam. Constant acceleration throughout whole period may not be desirable.

Acceleration during cam control may be high, while acceleration during spring control should be as low as possible in order to use light springs,- spring pressure must always overcome inertia of valve and parts with some margin.

To prevent exhaust valves being sucked open, minimum spring tension must be sufficient to resist suction in cylinder,- at least 11 pounds per sq. in. of valve area.

With moderate-speed engines, extremely high rates of acceleration are not needed when under spring control. An acceleration of 800 to 900 ft. per second per second permits favorable valve openings with engine of 15 to 20 B.H.P. at 2000 r.p.m. or less. Usually for such engines the provision of a spring of sufficient strength to prevent opening of valve by piston suction will meet the case.

Highest permissible rate of acceleration while valve is under cam control depends on type of follower. With roller type, whole load is taken or roller pin, - small projected area and poor lubrication; changes in surface velocity and inertia of roller cause skidding when pressure is a maximum; increased

weight. Should not be used on very high speed engines.

With curved slipper type, no overloaded bearing; small area of rubbing surface, increased wear; greater skidding on single face; may have greater radius and width of slipper than for roller.

Mushroom type in some ways the most satisfactory, but necessitates larger base circle on cam. Follower should be offset sideways to distribute wear over whole surface. Impossible to use low rate of positive acceleration, but safe to use high rate of acceleration for high pressure occurs at flank and not at tip of cam.

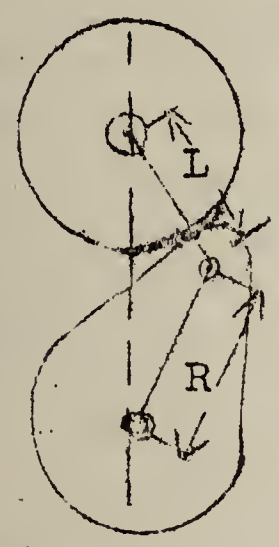
Desirable to keep base circle of any cam as small as possible in order to reduce rubbing velocity.

"Effective" radius of any cam is actual radius from center of camshaft to center of roller. Hence, it matters not whether cam be large and roller small or vice versa.

Cam Diagrams

1. Valve Movement diagram (equal time basis)
2. Valve opening diagram (for piston positions)
3. Velocity Curve.
4. Acceleration Curve.

Adjustment of Proportions. Certain adjustment of positive acceleration with tangential cam may be made by changing distance from center of camshaft to center of roller, - acceleration varies directly in proportion. Such change has opposite effect on negative acceleration on nose of cam. If pressure of spring balances inertia of valve parts at maximum speed, nose of cam systains slight pressure; at low speeds pressure on nose must be less than spring pressure, - short curvature on nose will not lead to undue wear. Curvature of flank and follower should be as large as possible, - small base circle and large roller.



With cams and followers formed of arcs of circles, velocity and acceleration of center of roller are the same as those for a piston having crank and connecting-rod lengths R and L.

Proportions of nose of cam should be such that $\frac{R}{L}$ is less than unity.

With mushroom follower, motion from flank is series of simple harmonic motions of varying amplitude. Radial velocities and accelerations about base and nose are proportional to distance from center of cam shaft to center of curvature.

Masked Valves. By recessing the valve, it is possible to start the motion of the valve earlier and finish it later. While keeping the time of effective opening and closing as before this greatly reduces the acceleration.

Effect on volumetric efficiency is the most useful feature of the recessed or "masked" valve, because it gives a large opening at bottom dead-centre and practically closes the valve before the piston has risen sufficiently to pump the charge back through the valve. A mask depth of about 1/6th to 1/8th of the lift is suitable.

Valve Springs. The function of the valve spring is to decelerate the valve moving parts during the latter part of the

valve opening and to accelerate them during the first part of the valve closure. In addition, the exhaust valve spring must be strong enough to keep the valve closed during the suction stroke when the engine throttled. During idling the piston suction may give a pressure difference of about 50 lbs. with a 2-1/2 in. valve. The spring pressure required for acceleration is sometimes in excess of that required to keep the exhaust valve closed during idling.

Cylindrical helical springs are employed almost universally, but occasionally conical helical springs may be used. In the larger engines two concentric cylindrical helical springs are frequently used.

Spring Force. To determine the spring force, - we must determine the maximum value of the deceleration; and, must design a spring capable of imparting to a weight equal to that of the reciprocating parts of the valve mechanism (including one-half the weight of the spring) an acceleration equal to the maximum deceleration determined by the cam outline for the maximum speed of the motor. The use of such a spring insures that the cam follower always remains in contact with the cam outline.

Following this procedure, and differentiating the equation for velocity we obtain the maximum; differentiating again we find the maximum acceleration.

For a Tangential Cam.

$$a = -\left(\frac{D^2}{(R' + r)} + D\right)\left(\frac{2\pi n}{60}\right)^2 \quad \text{in. per sec.}$$

$$\text{Spring force} = F = \frac{W N^2}{35200}\left(\frac{D^2}{R' + r} + D\right)$$

where R' is the radius of roller,

r, the radius of arc at end of cam,

and D, the distance from centre of r to centre of cam shaft.

N = r.p.m. of cam shaft.

Example:

For Mushroom Cam. The equation for the spring force becomes

$$F = \frac{W \cdot D \cdot N^2}{35200}$$

Example:

For Constant Acceleration Cam.

$$a = \frac{\ell \pi^2 n^2}{5400 f \alpha^2}$$

where f = the period of total time of lift in which deceleration takes place, and α = the angular velocity of the cam for the lift expressed in radians.

If we let α = the angle of lift in degrees

$$\text{Spring force} = F = \frac{W \ell n^2}{5.36 f \alpha^2}$$

Example:

In the case of the tangential and mushroom types of cams, the spring pressure obtained by these formulae are those which the spring must exert when the valve is fully lifted.

In the case of the constant deceleration cam the spring must exert a pressure equal to that given by the formula when the valve has attained its maximum speed and is beginning to decelerate.

Spring Formulae. The maximum safe pressure of coiled wire springs, and the deflection under a given load may be obtained as follows:

$$F = \pi \frac{Sd^3}{8D_1}, \text{ and } d = \sqrt[3]{\frac{8 D_1 F}{\pi S}}$$

$$C = \frac{8nPD^3}{E d^4}, \text{ and } n = \frac{C E d^4}{8 P D_1^3}$$

Where F = maximum safe load

C = the compression of the spring = lift

D_1 = mean diameter of the coil

d = the diameter of the wire

n = the number of coils in the spring

S = the maximum safe fibre stress

E = the torsional modulus of elasticity

and P = load in pounds.

S may be taken from 40,000 to 50,000,
E, 12,000,000 to 13,000,000.

Example:

It is observed from formulas, that for a given maximum pressure the stress to which the spring material is subjected varies directly with the diameter of the coil and inversely as the cube of the diameter of the wire. On the other hand, for a certain deflection per pound of load the number of coils varies as the fourth power of the diameter of the wire and inversely as the cube of the diameter of the coil. Combining these two propositions, - increasing the diameter of the coil increases the stress and decreases the number of coils, while increasing the diameter of the wire decreases the stress and increases the number of coils required.

<u>Selection of Springs.</u> Valve Springs must be considered from four points:

(1) The force at various points in the lift
(2) The maximum stress in the wire.
(3) The stress range from max. to min.
(4) Periodic vibrations in the spring itself.

(1) Force in various points in lift may be calculated, or a force-lift graph may be drawn from the displacement and acceleration diagrams.

The minimum force must be sufficient to hold the exhaust valve closed when throttle is almost closed; and, to avoid unnecessary loading on the valve gear, the maximum should be just high enough to allow a safe margin beyond the highest engine speed.

(2) The maximum stress in the wire must be well below the yield point, otherwise permanent set will take place.

Not advisable to exceed 60,000 lbs. per square inch, preferable to keep within 40,000 to 50,000 lbs. per square inch maximum.

(3) The stress range, i.e., the difference between the initial stress and the maximum stress should be kept down in order to avoid fatigue and rapid deterioration of the metal in the spring. The range should not exceed 24,000 to 30,000 lbs. per square inch according to the quality of the steel and the life expected.

(4) Periodic Vibration. If the mass of the spring itself be too great in relation to its stiffness (which is proportionate to its rate), the natural period of oscillation of the spring becomes large and may approach the period of the valve motion. Serious vibrations may then be set-up, and the spring is liable to fatigue and may allow the valve-gear to jump the cam.

The number of free vibrations per minute of the centre of the spring when the ends are held is:

$$N = 513\sqrt{\frac{R}{W}}$$

where N = number of vibrations per minute,

R = rate of spring, i.e. lbs., required to compress it 1 in. axially.

and W = the weight of the spring in lbs.

If it is found that N is equal to the revolutions per min. of the camshaft or is a simple multiple of it, it is practically certain that the spring will shudder and cease to function properly.

The spring found by calculation should be considered under heading (4) before being passed as suitable.

It will be apparent that (3) and (4) are in opposition, for if the stress range be kept <u>too</u> low, the spring loses stiffness and may develop slow vibrations in time with the valve gear. There is no practical difficulty, however, in satisfying both points in very high-speed engines

<u>Example:</u>

Valve Gear Details. Valve spring retainer is usually a cupped disc with turned down flange to center the spring, fastened to the valve stem in various ways. When the cam acts directly on a flat cam follower on end of valve-stem the follower may serve to hold the spring retainer.

Illustrations.

Rocker arms are usually pivoted in plain bearings; ball or rolling bearings are sometimes used.

Adjustment is usually provided for the tappet clearance. The amount of clearance should exceed the expansion of the valve stem, and depends mainly on the length of the valve stem; inlet valve requires less tappet clearance than the exhaust valve; in the Liberty engine the inlet clearance is 0.015 in., the exhaust 0.020 in. when the engine is cold. Too much clearance is to be avoided, as it produces noise and possible breakage of valve parts.

Illustration of forms of valve guides, push rods, tappet rods. Enclosed valve mechanism.

Location of Valves. Overhead valves permit the simplest form of cylinder head and water jacket, shortest and most direct passage of the gases, and, with overhead camshafts, a considerable simplification in the valve gear. The valves may either be seated in a flat head, or the head may be domed, in which case the valve stems are inclined to the cylinder axis.

Valves. Valves are of the poppet type with bevelled seats. The valve face is usually wider than the seat on which it closes. The width of the valve face must be small to ensure gas tightness. Seat is usually conical, 45° or 30°. Flat seats.

With conical seats the lift of the valve must be greater by

$$\frac{1}{\cos \text{ of seat angle}}$$

A characteristic of the poppet valve is its freedom from deterioration under the action of heat.

Valve must be cooled as much as possible. Cooling is accomplished by radiation from the head to the fresh charge, by conduction through the stem and by conduction through the face to the seat.

Face contact the most important.

Numerous ways of making poppet valves. Steel valves. Drop-forged in a single piece; but type most generally used consists of an alloy-steel head electrically welded to a carbon-steel stem.

Valve lift should be sufficient to give a maximum area under the valve in excess of the port area; a high average opening with respect to the size of the port is obtained which permits the use of a small valve. Decrease in size of valve head assists in the cooling.

The pressure drop through the inlet valve affects the pressure at end of suction stroke, and, thereby, the volumetric efficiency.

To obtain a larger valve area, multiple valves are sometimes used.

If lift is kept the same as for a corresponding single valve, the gas flow through dual valves will be much increased, - diameter of dual valves can be reduced, without decreasing the gas flow below that of a single valve. The valve area presented to gas pressure will be less, and weight of the two valves will be also less than that of a single valve. The power required to operate is les valve cooling will be better because the circumference of the two valves is greater than that of a single valve, and the distance travelled by the heat is less;

The main problem of the exhaust valve is heat dissipation.

Increase of exhaust valve area by increasing its diameter results in increased temperature of the valve. The heat received by the valve is approximately proportional to the square of its diameter, while the area of the heat abstracting seat is proportional to the diameter. Increased valve diameter gives increased valve temperature and liability of warping.

In addition to providing the maximum free area through the valves, the form of gas passage should offer minimum resistance, and permit smooth flow without formation of eddies, especially for the inlet valve. With a trumpet heat the gas-flow lines are improved, Approach of gas to the valve is usually smoother when the valves are inclined than when they are vertical

Valve Design. In the design of the valves the objects are:

(1) To provide the freest possible entry and exit for the gases.
(2) To keep the valves as small as possible, consistent with the first condition.

It is essential to make the orifice coefficient of the valve and its surroundings as high as possible, in order that it shall pass the maximum weight of gas with the minimum of pressure difference.

(1) The flow of the gases on either side of the valve port should be as free as possible, no abrupt bends nor changes of section.
(2) The lift should be equal to one-quarter the port diameter if possible.
(3) The angle of the valve seat should be fairly flat, 30° appearing to be good.
(4) The under side of the valve head, particularly in the case of the inlet valve, should be well radiused.

Use of unduly large valves, particularly large valves with reduced lift, should be avoided for the following reasons:

(1) Relatively large valves are invariably detrimental to compactness of the combustion chamber.

(2) For a given frictional resistance, and volumetric efficiency and pumping loss, a higher entering gas-velocity may be employed when the valve is of moderate size with a high lift, and well stream-lined, so that the degree of turbulence and therefore the poer output and efficiency are greater.

(3) Since the valves get rid of most of their heat through the seats, the larger the valve the higher will be its temperature. It is important to keep the temperature of the valves as low as possible, - the exhaust valve because its durability is a function of its working temperature, and the inlet valve because the entering gases are only too ready to take-up heat from the inlet valves and so penalize the vdumetric efficiency

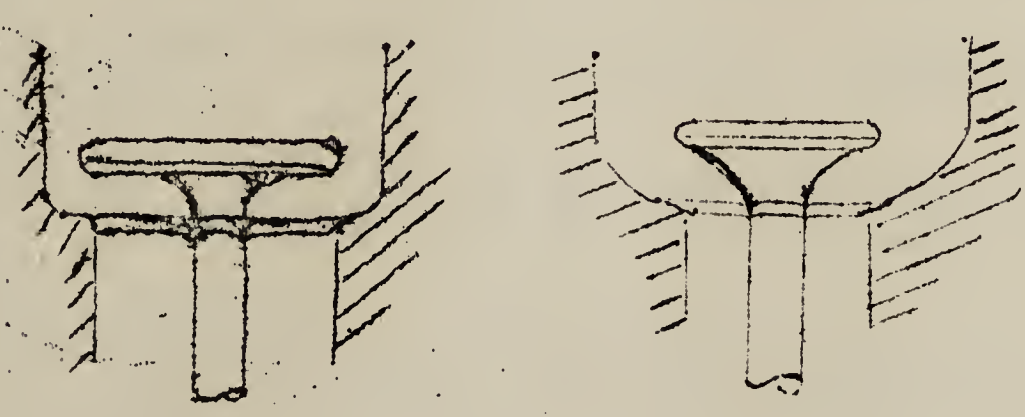
(4) Large diameter valves, particularly exhaust valves, result in unnecessarily heavy stresses on the valve-operating gear. That the valves should always be as light as possible is obvious, but there is a tendency to carry weight cutting in the valves al-

together too far, with the result that stretching, distortion, and overheating are liable to occur. The weight of the valve head alone, forms a small proportion of the whole reciprocating mass.

Since the valve gets rid of its heat largely through its seat, the seat should be made fairly wide. There is little doubt that the use of wide seats tends to a higher and better sustained efficiency.

The figures show two actual designs of valve in the full open position and the surrounding walls. The smaller valve is the more efficient owing to its better stream line flow and more efficient orifice coefficient; although the head is more massive, it is for the same weight of gas delivered the lighter of the two, and will have a longer life.

Number of Valves. In order to keep the temperature of the valve heads as low as possible and so increase the durability of the exhaust valves and the volumetric efficiency, it is, some-



times preferable to duplicate the valves rather than to resort to the use of very large sizes. In Ricardo's opinion, no single valve should deal with more than 200 lb. of air per hour as an approximation. When the valves are placed vertically in the cylinder head, duplication of the valves introduces no difficulty. In some cases it is desirable to use two inlet and three exhaust valves.

Two small exhaust valves and a large one may be used, the smaller valves being opened earlier, in order to get rid of the bulk of the high-pressure, high-temperature exhaust products before the large valve opens, and so to relieve this valve and its operating mechanism.

Cylinders

Considerablecvariety in the design. All in accord in adoption of overhead valves.

Cylinders are formed either singly, or in blocks of two, three or four. The single cylinder is flexible and permits freedom of movement without putting strains on other parts of the engine. Single-cylinder construction used in majority of engines and always in engines whose cylinders are all steel. The block arrangement with common jacket may result in slight decrease in weight of the jacket but usually an increase in weight of water, and may give inferior circulation. Main advantage of block construction is that it permits closer spacing of cylinders and diminishes over-all length (and weight) of the engine.

Cylinder Heads. Highest thermal efficiency is obtained for combustion chamber shape giving least ratio of surface to volume,- greatest mean indicated pressures are obtainable from such shapes owing to minimum heat loss to the walls occuring during the maximum temperature period.

Arrangement of combustion head and valves, a compromise between conflicting requirements.

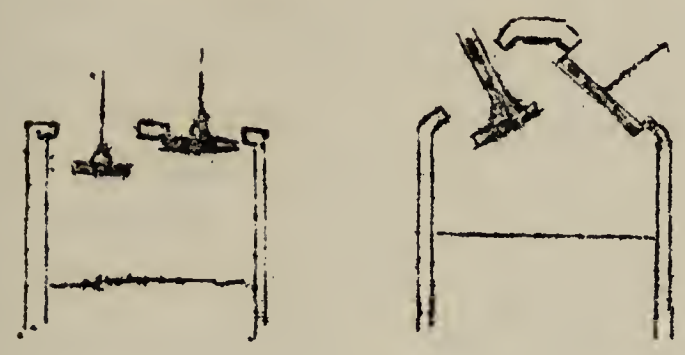
1. Thermal Efficiency,- The combustion head and valve disposition should be such as to present the smallest possible surface to the gases.

2. Volumetric Efficiency.- The valves should be of such size that they do not give excessive gas velocity through them, and the area of the passage leading to them should be greater than that of the valve opening. Gas passages should be free from sharp angles or sudden changes of area, and disposition of valves should be such that the pipework leading to and from the cylinder will satisfy the same conditions. The valves should be so arranged that free flow of gas is not restricted by too close proximity to the walls.

3. Mechanical Strength and Dissipation of Heat. Combustion head must be designed with due regard both to strength and rapid dissipation of heat, and disposition of valves must not be either weaken the head or obstruct the free flow of circulating water. The thickness of metal should be as nearly uniform as possible

4. Cost of Production.- Combustion head should be designed in such a way as to keep-down the cost of manufacture, and valves should be arranged so they can operated in possible manner.

Nearest practical approach to ideal shape shown in sketch,



B; while sketch, A, illustrates a fairly popular inexpensive arrangement.

Fulfills first and second conditions fairly well, but, for very high speeds it has the disadvantage that it is not possible to fit valves large enough for the best volumetric efficiency, unless the top of the cylinder be enlarged, which is detrimental. From mechanical point of view it weakens the combustion chamber and interferes with free flow of circulating water;

B is probably the best from the point of view both of thermal and volumetric efficiency, and is frequently employed. It does not weaken the combustion head, nor interfere with water circulation to the extent as A. Its principal objection is its cost.

In the Hispano-Suiza and Napier engines the cylinder head is flat of same diameter as barrel. With two equal valves, their maximum external diameter is half the cylinder diameter minus half the thickness of the bridge between them. This size of valve is frequently insufficient to give a desired opening. Heads with sloping sides without enlargement of diameter are used on the Curtiss OX and Mercedes, or with enlargement of diameter on the Liberty, Austro-Daimler, Fiat and others. Another way of meeting the difficulty is by the use of multiple valves.

Influence of a turbulent charge should be taken into account. Turbulence may be produced either by a high gas velocity past the inlet valve or by a flow of the charge during compression .

<u>Detachable Heads</u>. The detachable head has advantages. Cylinder boring, reaming and grinding are facilitated. It usually makes possible the use of larger valves and permits them being placed nearer together. Work of grinding valves and scraping out carbon is lightened.

Chief objection is the gas-joint and water-joint. With careful design and ample rigidity, the detachable cylinder-heads has much to recommend it.

Water Jacket. The design of the water jacket has much to do with the success of the engine in service, for uniform cooling is necessary. To obtain good circulation and to eliminate steam pockets, the water should enter at the lowest point in the jacket and leave at the highest point. A free path must be provided, especially about the valve ports, the spark plugs, and the gas passages to ensure a steady flow. No short circuits are permissible,- they cause excessive cooling at some portions and overheating at others. The valve springs should rest upon the walls of the jacket, rather than upon the port walls.

Location of Spark Plug. Form of head sometimes presents difficulties in location of spark plug. It should not be subjected to oil spray from over-lubricated piston nor to the heating effect of exhaust gases.

The position of the spark plug is determined by several factors - shape of combustion chamber, arrangement of the valves, compression ratio, and accessibility of the plug. It should be as near the center of the gas as possible. With high compression it is important that the spark plug boss be well water-jacketed.

Fouling of the plug may be minimized by placing it out of the path of the lubricating oil from the piston and of the wet constituents of the charge. Ample clearance must be provided to facilitate its removal and to prevent grounding. Multiple spark plugs help to insure the inflammation of the charge.

Cylinder Materials. Most important factor in determining the design of cylinders is the material used.

The function of the cylinder requires a material capable of withstanding the high temperature and pressures of combustion without excessive deformation and capable of providing a good bearing surface for the piston; in addition to these it should be comparatively light for its strength. Of the metals used none possess all the desirable characteristics. Cast iron, falls short in the matter of strength with low weight, and cast aluminum does not possess good wearing properties. Machine steel is used where low weight is necessary.

The following constructions are in use:

(1) All cast iron, with cylinders either single or in blocks. This construction leads to excessive weight.

(2) Cast-iron barrel, head, and part of jackets, but with aluminum sides to the jacket. For block construction only; it reduces weight and makes inside of jacket accessible.

(3) Cast-iron barrel and head, but with sheet metal jackets, either copper, steel or monel-metal.

(4) Steel barrels, aluminum heads and jackets. The steel barrel may be integral with its head or only a cylindrical shell. The aluminum jacket may be complete or may use the steel barrel as the inner wall.

(5) Steel barrel, mcast iron head and steel jacket.

(6) All steel. This construction is used more than any other. The cylinder is either machined out of a solid forging or made from steel tubing. The jackets are commonly stamped to shape in halves, and welded together and to the cylinder. The valve ports and guides offer some difficulties as compared with cast cylinder heads.

The design of a cast-iron cylinder that is satisfactory in both production and service requires good knowledge of foundry practice, of machine shop methods, and of the conditions of operation in the field. The metal must be properly distributed.

The main consideration is surface hardness, and particularly important with aluminum pistons lapping can be resisted only by surface hardness. Cylinders finished by grinding seldom wear so well as those which have been reamed.

With steel liners shrunk into aluminum cylinders the effect of the difference of expansion of the steel and aluminum tends to cause separation of the liner and casing when high working temperatures are attained.

A mild steel, case-hardned liner affords an excellent wearing surface. Its use is possible only when the liner is a plain open-ended tube of nearly uniform thickness and perfectly symme-

trical.

As regard material for built-up steel cylinders a high-carbon steel gives good results as regards wear, but it is rather difficult and expensive to machine.

Of the materials for cylinder liners, case-hardened steel is, in Ricardo's opinion, the best from the point of view of wear and the surface it attains; cast iron, if fairly hard, and reamed is probably second best, with high-carbon steel third. Low steel is quite useless for cylinder liners unless case-hardened. Bronze has been tried; it rise is impractical, owing to lack of surface hardness.

Valve Seats. Valve seats are integral with the heads in steel cylinders but have to be cast, or otherwise fastened, into aluminum heads. With integral or non-detachable valve seats it is impossible to take out the valves without taking off the cylinder or taking-out the piston. Detachable valve cages have been used but these result in imperfect cooling of the valve seat.

Valve ports and valve stem guides in all steel constructions are integral for each valve and are welded to the cylinder head. The guides are provided with bushings of steel or bronze; the valves must work freely in the bushings and give no leakage of air or exhaust gas between valve stem and guide.

Thickness of Walls. Several empirical formulae have been in use for calculating cylinder wall thickness.

$$\text{thickness} = t = \frac{\text{Diam. X Pressure}}{\text{2 X working stress}}$$

For cast iron. $t = 0.045$ Diam.
For steel $t = 0.018$ Diam.

No allowance need be made for wear or reboring.

The clearance space alone is subjected to the maximum explosion pressure.

Lowest part of the cylinder need not be more than one-half the thickness of upper end. Cylinders are sometimes turned of diminishing thickness from head to crank end; or upper part of the cylinder is reinforced by stiffening ribs.

Thickness of the Cylinder Head is determined mainly by considerations of stiffness; it is essential that the valve seats be free from deformation. With integral cylinder head backed by an aluminum casting, thickness cannot be reduced, because heat transfer is poor and the exhaust valve seat must have sufficient thickness to conduct the heat away and prevent warping.

Length of Bore. The piston should overtravel the cylinder bore. Length of cylinder bore will be the length of the piston plus the stroke minus the two over-travels.

Method of Attachment to Crank Case. Best method seems to be to tie the cylinder by through bolts to the main bearing caps on each side of the cylinder. This method of attachment avoids distortion of crank case

Cylinder Designs. In aircraft engines it is necessary to provide some form of composite cylinder construction, because the limitations of weight. Much diversity of opinion as to relative merits and demerits of the different built-up forms in use.

Illustrations

Cylinder construction used in 260 B.H.P. German Mercedes engines such as were fitted to the large bombing planes. High-carbon steel barrel screwed into a pressed or cast-steel head and welded; a light built-up sheet-steel jacket welded over all. Requires a specialized plant and special skill in welding. Long and well-cooled guides for the exhaust valves are a good feature.

Form of construction used in early Austro-Daimler, and later in 120 and 160 B.H.P. Beardmore six-cylinder engines. Cylinder barrel and head are cast in one piece of cast-iron, but inlet valve is fitted in a separate detachable housing. The lower end of the cylinder barrel is threaded to receive a steel flange for the holding-down bolts. Water jacket is of electro-deposited copper. Experience and precaution are needed to ensure a uniform deposition of copper and thorough adhesion to the cast-iron.

In later Austro-Daimler engine the cylinders are all steel with pressed-steel jackets and twin inlet and exhaust valves in the cylinder heads. One inlet valve-seat is removable Thickness of cylinders tapers from the top to the middle and increases again at the bottom.

Form adopted by the Maybach Company for their 300 B.H.P. six-cylinder engines used in the later Zeppelin airships and, in many of the larger aeroplanes. In one form of construction

the cylinder head, together with the whole of the water jacket, is of cast-iron, and a high-carbon steel barrel is screwed and sweated into the head, the lower end of the jacket being sealed by means of a rubber ring. In another form, only the cylinder head and the upper portion of the jacket are of cast-iron, the jacket of the barrel being a light seamless-tube also screwed to the cylinder head in same manner as the liner.

Another form of composite steel and cast-iron construction is used in the Maybach engine. The steel barrel is screwed with a buttress thread into a cast-rion head and comes up against a soft-brass washer. Water jacket is a machined forging and screws on to the cylinder head where it is sweated-in. The valve-stem guides have cast iron bushings.

Construction used by the Benz Company. The whole of the cylinder barrel, cylinder head, and holding-down flange are cast in one piece in cast-iron over which a light pressed-steel jacket is electrically welded direct on to the cast iron. The welding is a difficult problem. Here, we have an example of single cast-iron cylinders seldom used. In Benz-230 engine the jackets are extended well down towards the base flange, and are provided with corrugations to take care of the expansion. The cylinder walls taper from the top to the base of the jacket. In Benz-300 engine the cylinder head is of cast-iron and the rest of the cylinder is of steel. There are parts for two inlet-valves and one exhaust-valve. The steel liner screws into the cast-iron head and makes a watertight joint by bedding into cement in small groove.

The construction shown is that adopted by Rolls-Royce. The cylinder barrel and head are forged in one piece, and the inlet and exhaust valve elbows are screwed and welded into place, the whole body being covered by pressed-steel jackets, welded over all. This is a form of construction which has found much favor; both the cylinders, themselves and the water-jackets are built-up out of steel by welding.

This all-steel construction is exemplified in the Liberty cylinder.

The cylinder and cylinder head as a plain thimble, to screw and weld the valve ports and valve guides into this, and finally to weld over all a light steel water jacket pressed from thin sheet and made up in two halves. This form is light and sound, and is not very expensive. The chief objections to it are:

(1) It is applicable only to separate cylinders.

(2) The water connections are attached only to light-sheet-metal jackets, which are liable to crack from vibration.

(3) Freedom of design is rather limited by constructional difficulties.

Cylinder construction used in the earlier Sunbeam engine in which complete blocks of cylinders were cast in iron together with their cylinder-heads and the upper portion of the water-jackets; the whole of the sides of the casting below the valve outlets are covered by light sheet- metal plates. In Sunbeam engines, such as the 200 B.H.P. Arab engine, the complete cylinder block was cast in aluminum with thin steel liners shrunk-in.

Construction of the Hispano-Suiza cylinder block in which a complete steel thimble forming both the cylinder liner and valve seats is screwed directly into an aluminum cylinder block.

The steady liner is threaded throughout the entire length of contact with the aluminum. The aluminum jacket is a block construction for four cylinders and completely surrounds the steel liners so that the valve ports go through both the aluminum and the steel. The valves seat directly on the steel thimbles. Perfect contact of the two metals is necessary at the cylinder-head as well as the barrel. No contact of water with the steel cylinder. The steel liners are attached to the crankcase by bolts. This form has advantages from a constructional point of view, but it is open to the objection that the heat has to pass through two separate thicknesses of metal, in contact only be screwing, before reaching the cooling water, - a much reduced margin of safety for dealing with excessive rates of heat flow.

In the Siddeley Puma engine, blocks of three-cylinder heads, together with the upper portion of the water-jacket, are cast in aluminum with pressed-in bronze valve seats. Into these are screwed, for a short length only, thin steel cylinder barrels. The liner has no head; the aluminum heads are cast in sets of three and the steel liners are screwed and shrunk into them. Thin die-cast aluminum water jackets in threes are bolted to the heads. The lower joint between the aluminum jacket and the liner is made by a screwed gland.

The Curtiss OX eightecylinder V-engine has separate cast-iron cylinders with brazed monel-metal jackets;

In later Curtiss K engines a steel-aluminum block construction is employed; the steel liners with integral heads are turned with stiffening flanges on the outside, with a packing retaining

flange at the bottom and a central stud at the top. The upper end of liner of enlarged diameter is threaded on the outside. The aluminum cylinder head is a block casting into which the six cylinders are screwed. To ensure intimate contract the stud is drawn-up by a nut. The hand-casting matches up with and is bolted to a flange on upper end of cylinder block which is integral with the upper half of the crankcase. Jacket water is in contact with the liner below the combustion space only. Water tightness is obtained by packing between the lower liner flange and the crankcase.

The Napier "Lion" engine employs liners with integral heads which are fastened to a four-cylinder aluminum-block head by four valve seats in each cylinder. These seats are screwed into the cylinder head. The barrel jackets are separate and of pressed steel, made in two halves welded together and to flanges on the cylinders at the top and bottom of the water spaces.

The form of construction in the Vauxall racing-car engine adopted by Beardmore in the large 750 B.H.P. six-cylinder aero-engine has much to recommend it. The cylinder block construction consists in carrying the aluminum crankcase up to the cylinder head and inserting in its loose steel liners machined all over and provided with flanges at the top nipped between the top of the crankcase and the cylinder head. This forms the water-jacket. It is both light and inexpensive. It renders the top half of the crankcase very rigid against bending or torsion. It is simple and easy to cast. The water connections can be secured to the main casting, and not to light sheet-metal jackets. The liners being of very simple and symmetrical form can be made from mild steel and case-hardened. To ensure the maximum of structural rigidity the crankcase is made deep, the cylinder water-jackets were cast in one piece and rigidly attached to form an additional girder, while through bolts extend from the cylinder block to the bottom of the crankcase, thus forming a structure of rigidity both as regards torsion and bending.

<u>Air-Cooled Cylinders</u>. Problems becomes different, weight is not longer the sole consideration, high conductivity being equally, if not more important. In engines with rotating cylinders cooling conditions are favorable; plain steel cylinders machined throughout from a single forging are usually used.

Clerget engine. The Gnome cylinder.

The Le Rhone cylinder. Whole of cylinder and head is machined from a single piece of steel, but a very thin cast-iron liner is pressed-in; construction is curious.

Illustration of air-cooled cylinders in machined steel and in cast-iron,

All-aluminum cylinders have been tried with fair success, but there is doubt of their durability; their considerable longitudical expansion increases tappet clearances and alters valve timing.

The satisfactory construction of an air-cooled cylinder depends on keeping down its temperature. With overhead valves it is essential to make the cylinder head of the best conductor.

R.A.E. cylinders. Cylinder of thick aluminum casting with steel cast-in valve seats and thin steel liner shrunk-in. Maximum permissible working temperature is below 300° C;

Contact between liner and cylinder body gradually failed, resulting in overheating of the liner. This form of construction abandoned in favor of an aluminum head cast-in to a plain ribbed steel barrel.

Tests show that the all-steel cylinder gives higher fuel consumption and lower brake power than does the aluminum-headed cylinder. With an aluminum capped steel cylinder impairs the contact after a few hours' running.

Cylinder distortion may arise from the cooling air blast being firected against one side of the cylinder.

With longitudinal fins and comparatively uniform distribution of air flow, the distortion is not noticeably less. The exhaust side will be the hottest, and the temperature will be less uniform than with circumferential fins and a free blast on the exhaust side. Furthermore, longitudinal fins do not stiffen the cylinder as strongly against distortion as to circumferential fins.

PISTONS

The piston serves a triple function, - forms a movable wall of the combustion space; receives and transmits gas pressures to the connecting rod; and forms a crosshead.

High degree of gas tightness is essential; - not easy to make piston gas-tight within the cylinder-bore. Fit of piston cannot be depended-upon to insure gas tightness; and flexible, split rings are inserted.

Pistons should be as light as is consistant with reliability and durability. The length of pistons used have varied much. Long pistons have the advantage of more surface to take the wear, but the disadvantage of greater weight. The short piston has been found to wear well, - the lighter it is, the less is the wear.

Broadly speaking, the main objects to aim at in the design of a piston for an air plane engine are:-

(1) To reduce friction to the lowest possible limit.
(2) To reduce the weight.
(3) To dissipate heat to the walls of the cylinder.
(4) To prevent the passage of oil into the combustion chamber.
(5) To provide adequate support for the wrist pin.

Conditions 1 and 2 are largely interdependent. The resolved component of the inertia forces exceeds the gas pressure; hence if weight is reduced the average bearing pressure is reduced; for same bearing pressure per square inch, and therefore for same durability, the area of bearing-surface may be reduced nearly in proportion to the weight.

Piston friction is dependent also on the nature and condition of the lubricating oil adhering to the cylinder walls. Conditions are different than for bearings; although the rubbing velocity is higher, the average load is slower and small. The area of surface and the viscosity of the lubricant play an important part.

Piston Friction. As a rule, piston friction constitutes by far the largest item of mechanical losses.

The cause of piston friction is probably the shearing of the oil film formed between the cylinder and piston; and the power absorbed is dependent upon the thickness of the film, the viscosity

of the oil, the area, and the speed.

The coefficient of friction between the cylinder and piston is much higher than for bearings, - probably accounted for:-

By reduced supply of lubricant as compared, and thickeneing of the lubricant.

If the supply of oil be profuse, a considerable proportion of the oil will finds its way into the combustion space. A certain proportion of the lubricant will be partially carbonized, and thickened; its viscosity and resistance to shear is thereby increased.

Desirable to keep the clearance between trunk of the piston and the cylinder walls down to the lowest possible limit consistant with leaving provision for expansion and distortion.

If an unduly large clearance is permitted -

Noise will be set-up due to the piston crossing violently from one side of the cylinder wall to the other. This may be serious. There will be tilting of the piston which is objectionable, as it causes the rings to slide in their grooves, and soninduces wear. When ring grooves become worn burning gases pass down between the piston and the cylinder walls at a rate sufficient to carry-off much of the lubricating oil and carbonize the remainder. Desirable to provide a vent forgases which pass the piston rings.

The piston and cylinder do not remain truly circular when heated, a certain amount of distortion always occurs. Distortion of the piston is generally due to the wrist-pin and wrist-pin bosses; evil effects can be largely avoided by cutting-away the sides of the piston.

Slackness of the rings in their grooves is a fruitful cause of passage of oil; at every reversal of the piston oil is pumped past the rings.

Clearance should be kept as small as is consistant with the distortion that may take place; evils attendant upon considerable clearance greatly outweigh the advantages. Portion above trunk may be allowed greater clearance,- its chief function is to maintain the piston rings in place; and since this portion is much hotter, greater expansion and distortion takes place; must not come in contact with the cylinder walls.

Piston friction is reduced as the temperature of the cylinder wall increases. Friction of the cross-head portion varies nearly as the square of the speed and other conditions; there is a certain amount of constant friction due to the pressure of the piston

rings, friction is dependent on the stiffness of the rings, and so long as the rings are free in their grooves it is nearly independent of the speed or other conditions. Much misapprehension as to proportion of piston friction due to piston rings, it does not amount to more than about 0.3 lb. per square inch of piston area, and is often even less.

Area of bearing-surface of piston has a considerable influence on piston friction; opinions differ. All that is accomplished by increasing bearing surface is to transfer wear from the piston. Question of bearing surface from point of view of wear is intimately bound-up with proportion which inertia pressures bear to gas pressures. Advantageous to reduce bearing surface to the lowest possible limit.

Piston friction can be sometimes reduced by drilling holes in the walls of the piston, and entrapped oil to escape.

Some designers consider it desirable to remove all inoperative surface in order to reduce the area of oil film in shear; relatively small area of bearing surface suffices. Pistons do not seize from overloading of the bearing surfaces; piston seizures are due to insufficient allowance for expansion or to distortion, and to complete failures of oil supply. The wear with cast-iron pistons is mainly due to piston rings. In the case of aluminum alloy pistons it is somewhat different, because of lapping effect.

Piston Weight. Piston must be as light as possible in order to keep down inertia forces, and must be thick enough in the head to conduct the heat away rapidly from the center to the circumference. With cast-iron pistons, reduction in weight has resulted in many troubles, especially in burning and cracking of piston head.

With cast-iron pistons, the total reciprocating weight may be 0.42 lbs. per square inch of piston area; with aluminum pistons this weight may be reduced to about 0.22 lbs.; per square inch of piston area.

Weights of modern cast-iron, steel, and aluminum pistons complete.

Diameter in ins.	Weight in Lbs.		
	Heavy Cast Iron	Steel (welded) of screwed	Aluminum: Short and Slipper Type
4	3.85	2.4	1.00
5	5.95	3.9	3.25
6	8.65	6.0	4.90

Dissipation of Heat. It is necessary to consider the heat dissipation.

The heat from the crown of the piston is disposed of -

(1) Through the rings to the cylinder walls.
(2) Through the bearing surface to the cylinder walls.
(3) To the oil and air below the piston.

There is evidence in support of the theory that the bulk of heat passes to the cylinder walls via the piston rings; heat can only be transmitted rapidly to the cylinder-walls through the skirt or bearing surface.

Proportion of heat carried-away by the circulation of the air and oil below the piston need not be taken seriously into consideration.

Most important consideration is transmission of heat from the center of crown to the circumference. In order to facilitate this heat transmission, the crown should be made as thick as possible consistent with the weight limitation, and the conductivity of the material should be as high as possible. Aluminum alloys have come into vogue for pistons; not only is their weight about one-third that of cast iron, but their conductivity is about five times as great, With such alloys the rate of heat transmission is so high that there is no need of making the crown of the piston any thicker than is needed for strength.

Since heat is dissipated largely by condition to the cylinder walls the center of the piston gets much hotter than the circumference. This large difference of temperature, and of expansion between the center and the circumference, introduces serious stresses. There is also danger that the center will become so hot as to cause preignition.

Results of experiments by Professor Cocker follows:

Temperature of	center of piston-head,	707° F.
" "	piston-head 3/4" inside of edge	550°F.
" "	Outer edge	465°F
" "	Inner surfade of cylinder liner	245°F
" "	Water jacket	200°F

Hopkinson on a Siddeley engine shows the cast-iron piston head temperature as over 900° F. With aluminum pistons this temperature is much less.

Heat received at the center of the head must pass to the circumference, and this should be provided for either by adequate thickness of metal from center to periphery or by provision of radial ribs which serve the double function of heat carriers and stiffening members.

Differences in Expansion

If we take the difference in the temperatures of a cast-iron piston head and the cylinder wall as 550° F., and with a coefficient of expansion of 0.000,00556, we have for every inch of diameter an expansion of

$$e = 0,000,00556 \times 550 = 0.003''$$

For a difference of temperatures of piston trunk and cylinder walls of 180° F. we have for every inch of diameter an expansion of

$$e = 0.000,00556 \times 180 = 0.001''$$

The piston should be stepped or tapered to meet this condition.

Example:

For aluminum pistons in cast-iron cylinder, if we take the increase of temperature of piston-head as 560° F. and that of the cylinder wall as 160° F., and with a coefficient of expansion of aluminum of 0.000,01234, we have for every inch of diameter an expansion allowance of

$$e = 0.00001234 \times 560 - 0.00000556 \times 160 = 0.006''$$

For an increase of temperatures of piston skirt of 275° F., we have for every inch of diameter of expansion an allowance of

$$e = 0.00001234 \times 275 - 0.00000556 \times 160 = 0.0025''$$

Example:

Passage of Oil into the Combustion Chamber. One common form of trouble with high-speed engines. Passage of oil past the piston rings is due to the following:

(1) Oil is forced-up against the rings on the downward stroke of the piston.

(2) The motion of the piston rings in their grooves tends to pump the oil into the combustion chamber.

To prevent the passage of oil into the combustion chamber, the following considerations should be taken into account:-

(1) The setting-up of a hydraulic pressure can be prevented by perforating the bearing surface of the piston, and also by venting the piston just below the bottom ring.

(2) As the piston travels upwards and downwards each ring functions as a valveless oil pump.

In order to reduce this pumping as far as possible -

(1) The rings should be made as close a fit in their grooves as possible.
(2) Venting should be provided below the lowest ring.
(3) The tendency further checked by drilling holes through the ring groove behind the lowest ring.

A widespread belief that passage of oil into the combustion chamber is dependent on suction in the cylinder; the theory is fallacious, for the actual quantity of oil passing the rings is a function of the speed alone. When an engine is running at or near full load, the oil passing into the combustion chamber is burned with the fuel; when running light, the flame temperature is insufficient to burn the oil and it accumulates.

In normal high-speed engine, about 90% of the consumed lubricating oil is burned in the cylinder as fuel. When determining the efficiency of an engine, the hourly consumption both in fuel and oil should be taken into account. In engines the consumption of oil is so high as to reduce the fuel consumption and so give a fictitious fuel economy.

Piston Material. Desirable features of piston metal are,- low coefficient of expansion, light weight, high heat conductivity and slight deterioration at high temperatures, resistance to wear, and ease of manufacture. Metals in common use are cast-iron and aluminum. Relative advantages of cast-iron and aluminum are so evenly balanced and their success is so much a matter of design and production methods that neither can be said to outclass the other by a wide margin.

Table giving characteristics.

Properties of Piston Metals

Metal	Weight lbs. per cu. in.	Melting Point Degree F.	Tensile Strength lb. per sq. in.	Coeff. of Linear Expan. per deg. F.	Relative Heat Conductivity
Aluminum	0.0924	1200	15000	0.00001234	3.8
Cast-Iron	0.2600	2300	20000	0.0000556	1.

On the score of light weight and high thermal conductivity, aluminum possesses a decided advantage, but in other features it is inferior. It is also more susceptible to abrasion and does not present such a good bearing surface to the wrist pin and the piston rings,

Strength, hardness, and coefficient of expansion may be modified by properly alloying and casting. A popular alloy contains between 6 and 14 per-cent copper; the remainder almost entirely aluminum with very little tin, magnesium and manganese and no zinc. A sand casting of this alloy shows an ultimate tensile strength of 20,000 lbs. per square inch of 480° F. and 12,000 lb. per square inch at 660° F. The addition of the copper brings the weight to about 0.107 lb. per cubic inch. The strength is increased by casting the metal in permanent molds. Another alloy, which is conceded to be most satisfactory is composed of copper 4 per-cent, nickel 2 per-cent, magnesium 1.5 per-cent, and aluminum 92.5 per-cent. The chilled casting has a strength of 25,000 lb. per square inch at 60° F. This alloy responds well to heat treatment with an improvement in physical properties.

The properties of cast-iron pistons are also benefited by the suitable addition of certain elements. The addition of 0.1 per cent of vanadium increases the strength from 10 to 25 per cent, toughens the metal and produces a uniformly fine grain.

A high-grade piston casting should have an ultimate tensile strength of 26,000 lb. per square inch. Semi-steel is sometimes used.

Sand-cast aluminum pistons have sometimes given trouble owing to porosity of the metal, particularly at the juncture of the hollow trunk and head; pistons have broken-away at this point. The defect has been remedied by casting the pistons in metal dies.

Heat Treatment of Piston. An important factor in piston manufacture is the reduction of the internal stresses in the metal resulting from shrinkage while cooling in the mold; is formed with a dry-sand core, as is the usual practice, this core may be so hard as to prevent the free contraction of the piston while the metal is solidifying and the walls will therefore be in a state of tension. On account of the stresses being unequally distribute if the piston is finished in this condition, it will gradually warp as the stresses are relieved.

"Normalizing" is hastened by suitable heat treatment. A satisfactory process for cast-iron is to take a roughing out over the exterior of the piston to a temperature of 800° F., hold for 30 minutes and allow to cool slowly in the furnace. Aluminum-alloy pistons are normalized either by immersion in oil which is held at a temperature of 475° to 575° F., by baking for several hours at the same temperature, or by annealing at 750° to 850° F.

In the 4 per cent copper-nickel-magnesium-aluminum alloy, the ultimate tensile strength may be increased 30 per-cent by proper heat treatment. After a protracted heating at 950° to 1,000 F. the casting is held at 900° F. for 48 hours in a nitrate bath, quenched in boiling water or oil, and finally tempered for 48 hours at 480° F. The resulting tensile strength is approximately 28;000mlbs. per inch at 60° F. and 23,000 lb. per sq. inch at 480° F.

Recently nearly all aero-engines have been fitted with pistons made throughout of aluminum alloys. The objections to an all-aluminum piston are:-

(1) Owing to the high rate of expansion with temperature rise, a large clearance must be allowed; this causes a knock when the thrust is reversed.

(2) Aluminum permits particles of grit to be embedded innit, and causes wear of the cylinder walls.

(3) Aluminum castings are liable both to grow and to distort, so that still more clearance must be allowed on this account.

In addition to reduction in weight, by substitution of aluminum alloy for cast-iron, the aluminum pistons carry a much lower temperature due to their greater heat conductivity.

The lower temperature of the aluminum piston reduces the rise in temperature of the incoming charge during the suction stroke and thereby increases the volumetric efficiency of the engine, and it permits the use of a higher compression ratio without danger of preignition and thereby increases both the capacity and efficiency of the engine.

An appreciable gain in power output and thermal efficiency is obtained by substituting properly designed aluminum pistons for the ordinary cast-iron ones.

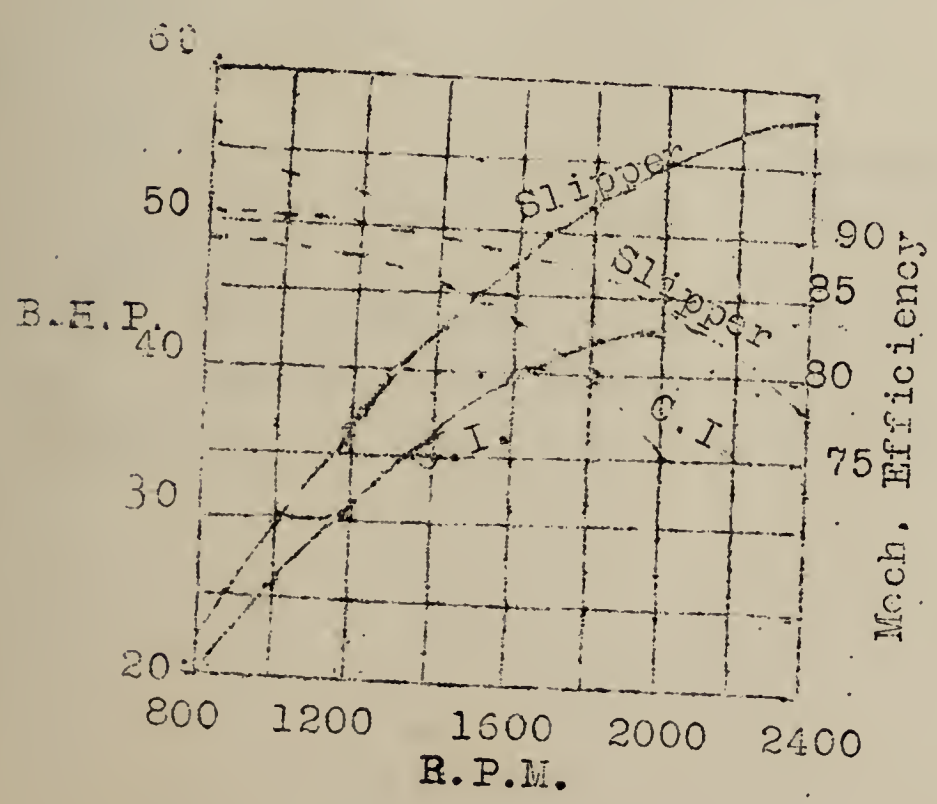
The increase in power obtained may amount to 15 per cent at the same speed. This improvement is due to the reduced piston friction, which is about 60 per cent lower and is also due to the lower total reciprocating weight, which is only about one-half of the cast-iron weight.

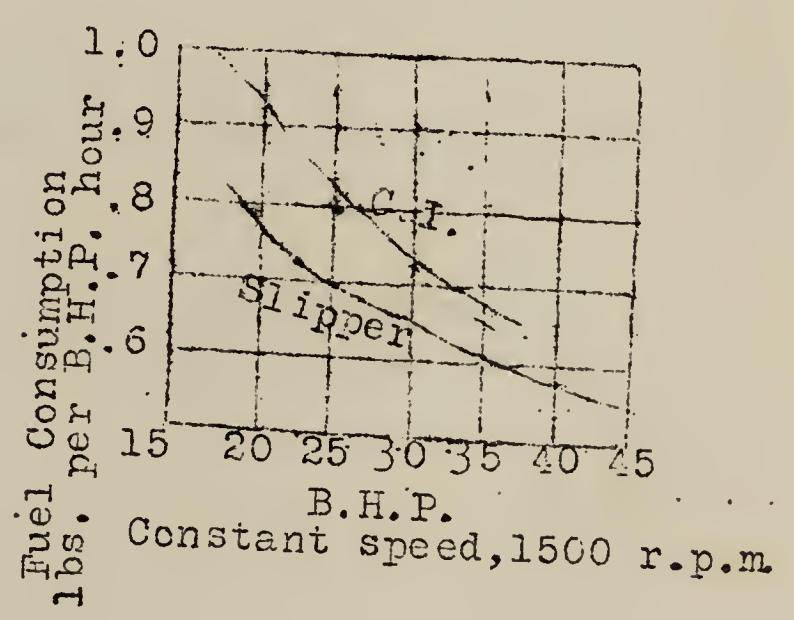
The improvement in the thermal efficiency is due to the better heat-conductivity of the aluminum, so that the surplus heat is better dissipated.

Typical Pistons.- Maybach, cast-iron, plain without ribs; Benz, cast-iron, thin head with hollow conical steel pillar riveted to head and resting on middle of wrist-pin; Benze, aluminum, ribbed construction; Hispano-Suiza, aluminum, thick walls; Curtiss OX, aluminum, thin-ribbed; Liberty, aluminum, without ribbing. Split pistons; Composite pistons.

Ricardo- Slipper piston. In order to reduce weight and to prevent distortion, it is desirable to transmit the pressure as directly as possible from the crown of the piston to the connecting-rod. In the design of the Ricardo slipper piston two main vertical transverse ribs transmit the load from the head to the wrist-pin. The sides of the piston have been entirely cut-away below the lowest ring, and the bearing surface is much reduced. It represents probably the lightest possible construction and it has been much used in very high-speed engines, where its light weight, low friction loss, and oil-resisting properties have rendered it of great advantage.

Diagrams giving the results of tests on a four-cylinder 4" X 5-1/2" engine fitted with cast-iron and slipper pistons.





The crosshead portions, or the slippers, of this type of piston in some constructions have been made of cast-iron, the two portions being bolted together or made to trunion about the wrist pin.

Piston Knock. Owing to large clearance with aluminum pistons, it is difficult to obtain quiet running. Various devices have been experimentedmwith in endeavour to overcome the trouble, - pistons of small normal clearance with slit skirts; springs between connecting-rod and piston; wrist-pin out of center in the piston; bearing surface severed from the crown.

None of the methods solved the difficulty of piston knock, and it is probable that where extreme quietness is required it is better to employ cast-iron for the bearing surfaces in all except very small pistons. With all-aluminum it is necessary to allow a clearance on the bearing surfaces of approximately 0.002 in. per in. of diameter. When the total clearance exceeds 0.005 in. to 0.006 in. piston knock becomes audible; all-aluminum pistons up to 3 in. can be made to run quietly, but above 3 in. diameter, it is extremely difficult, to ensure quiet running.

Piston Dimensions.

Piston-head thickness. Thickness = $t = \frac{d}{4} \frac{P}{S}$

Where P is the maximum gas pressure and S the safe working stress.

For cast-iron pistons,

$t = 0.075\ d$, at the center.

$t = 0.1\ d$, at the barrel.

If the piston is ribbed.

$t = 0.06d$

where d is the inside unsupported diameter.

For aluminum pistons, the thickness would be about 20% greater.

The thickness is for strength only.

Increased depth of metal at the walls is of advantage not only for strength but also for improved heat conductivity.

Several empirical formulae for thickness of piston heads; they are of little value.

Piston Pin. The form and dimensions of the piston are determined to no small extent by the piston pin.

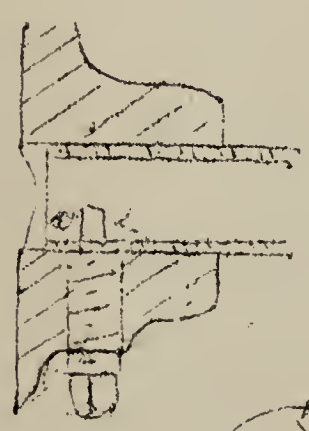
To calculate dimensions of the piston pin we would consider the pin as a bearing journal and, as a beam supported at both ends and loaded uniformly.

A hollow pin is preferable to a solid one;

The pin may have a running fit in either the piston or the connecting rod or it may float in both. Since it receives the gas pressure and the piston inertia forces through the piston bosses and transmits them through the rod, it must be of liberal diameter to prevent rapid wear; the diameter should be sufficiently large to reduce deflection of the pin to a minimum.

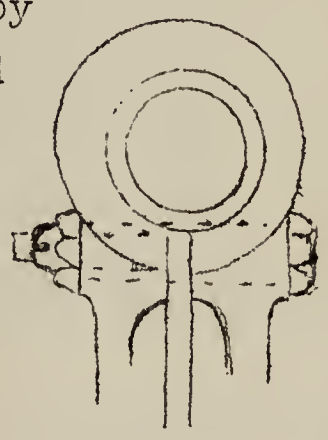
The bore of the pin is sometimes increased from the center to the ends for lightness.

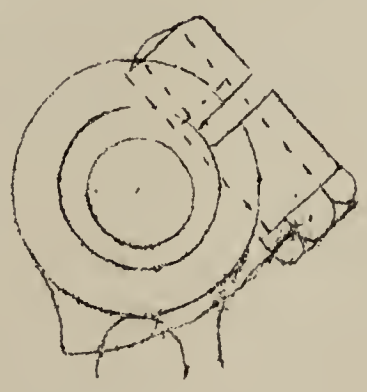
If the pin is held fast in the piston it is assumed that a tight fit in the bosses would be equivalent to fixing the ends as beam, a pin designed on this basis will gradually work itself loose in the piston.



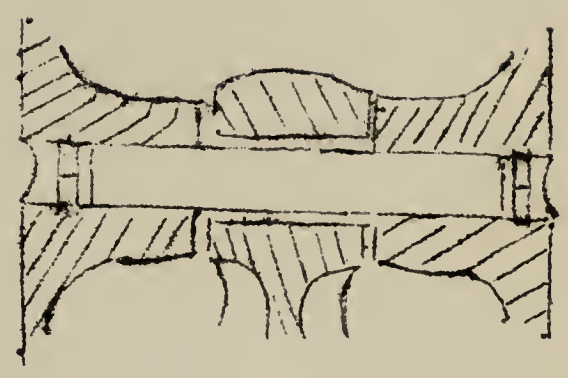
Various methods of holding the pin in the piston-bosses. One end only is held.

When the pin is held by the rod the connecting-rod end must provide a rigid anchorage.





Methods of holding a full-floating wrist-pin. The wrist pin bearing should never give trouble, nor show measurable wear, provided the pin is stiff enough to resist deflection and is adequately supported. Trouble with this bearing is almost invariably due to deflection of the pin.



Although the pressure on the wrist-pin bearing is high, the rubbing velocity is low, and the average load factor is not high. If the pressure is distributed uniformly over the surface of the bearing, the rate of wear and the risk of seizure should be very small. Although the bosses may be rigid their attachment to the piston is not. Deflection of the wrist pin under maximum pressure is frequently excessive and quite sufficient to concentrate the loading on the two extreme ends of the bearing. It is of utmost importance either that the wrist pin shall receive its load from the piston crown at points as near its center as the connecting-rod bearing will allow, or its diameter shall be such that, when loaded at the two extreme ends there is no appreciable deflection. For engines of normal compression, the diameter of the wrist-pin should be at least one quarter of that of the piston. Unless the pin be of large diameter, it is useless to provide a wide bearing at the connecting-rod. In Ricardo's opinion it is doubtful whether any use can be made of a bearing width exceeding 0.35 of the piston diameter. Tendency to wear the pin oval can be overdome by use of a floating wrist-pin which permits the pin to turn freely both in the connecting-rod bearing and in the piston bosses. A floating wrist pin removes difficulty of fixing it in the piston.

Even with full floating wrist pins in an aluminum piston, the aluminum bosses expand and the pin becomes free.

Any play at the wrist pin gives rise to a knock at the sudden reversal of pressure at the end of the stroke. When knocking once begins it increases rapidly.

For the material of the wrist pin surface hardness should override all others, case-hardened steel should give a glass-hard surface.

Piston Rings. Piston rings are made of cast-iron fully machined, exactly ground to size on the edges and outer curved surface, and usually peened on the inner curved surface.

Three rings are commonly used, but four rings are occasionally seen; also, two narrow rings have sometimes been placed in one groove. A scraper ring has sometimes been employed near the bottom of the skirt.

Piston rings have three important functions. (1) They form a seal against the escape of gas between the piston and the cylinder. (2) They constitute the principal path of heat transfer between the piston head and the cylinder wall, and (3) they prevent the passage of oil from the crank-case into the combustion chamber.

To perform these duties the rings should present a continuous unbroken surface to the gase in the cylinder, they should bear firmly and uniformly against the cylinder walls and they should fit closely in the grooves.

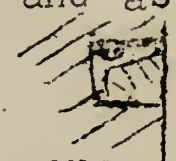
Three paths are open to the flow of gas or of oil past the ring: (1) the gap where the ring is split; (2) the contact face which engages the cylinderwall; and (3) the clearance between the ring and the groove. The gap is negligible in the volume of gas or of oil it may pass. The accuracy of fit of the bearing face of the ring against the cylinder, and the uniformity of the bearing pressure have great effect upon leakage. When the piston is new the clearance in the grooves may be as low as 0.0015 in.

Eccentric Rings, gives narrow side surface against the grooves and a greater space under it at the cut making it less gas tight. Concentric rings are most used.

Rings fitted by peening. Top ring with small slots in its top side, so pressure gets into the ring groove, adds materially to friction and wear. Ring pressure should be as small as can be made and secure gas tightness.

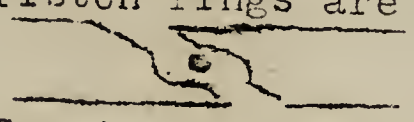
Cross Section. Cross section of rings is usually rectangular; sometimes modified by cuttings. In such cases, width of the bearing surface is narrower than the ring, and the unit bearing pressure against the cylinder is increased.

Special Forms. Ring beveled on one side, and assembled with a steel spring, behind it;



T.N.T. Rings with initial torsion or twist. Compound Rings; the "Leakproof" ring, one ring has an outward flange and the other an inward flange.

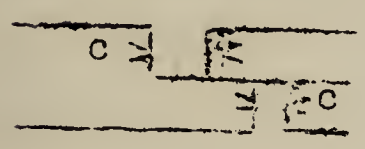
Piston rings are seldom pinned to prevent rotating; change of rings rotating is small.



Gap is usually made by a diagonal cut at an angle of 30° to 45°; stepped or lapped ends sometimes used.

Necessary gap when cold maybbe calculated.

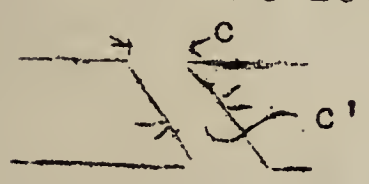
Difference of temperature of the upper ring and the cylinder wall would not be more than 300°; and not more than 200° for the lowest ring for cast-iron pistons. The width would be c =



$$\pi B \times .000,00556 \times 300.$$

Example.

For the lowest ring, $c' = \frac{2}{3} c.$



If diagonal cut is made in the ring, the cut could be less, depending on the angle of the cut.

With aluminum-alloy pistons the clearance may be reduced to about two-thirds this figure. The diagonal cut is most extensively used.

Special and patented forms of ring joints. Does not make much difference what this form of joint is; but the cut should be of sufficient width to prevent the ends from coming together under the variations of temperature and it need not be more than that amount. Placing cuts evenly spaced over the circumference is not best arrangement.

<u>Piston Ring Pressure.</u> Unit pressure between ring and cylinder lies between the limits of 5 and 6 lb. per square inch; the higher the pressure the greater the friction loss and the more severe the pounding upon the ring-groove walls.

The size of the ring to exert the required pressure may be calculated by the principles of the elastic curve of a uniformly loaded beam. Conditions for uniform stress cannot be realized. Size should be checked to see that fiber stress is safe, both when in place and when being placed in the piston. When sprung over the piston

$$t = B\sqrt{\frac{S}{E}} = \frac{B}{32}$$

Stresses in Piston Rings. Rings of good iron will return to their original shape after being stressed nearly to the breaking point. The ultimate flexural strength may reach 75 percent above ultimate tensile strength under axial load,-allowable fiber stress may be greatly exceed that permitted in direct tension.

Width of Ring. A narrow ring is preferred to a wide one in order to reduce friction; and a concentric ring is preferred to an eccentric ring in order to give better bearing in the piston grooves. Avoid use of delicate parts that may be broken in service or in assembling the ring to the piston.

Choice of Ring. Use ordinary plain concentric cast-iron rings. Rings should be made as narrow as possible; the narrower the ring the less both the friction and the inertia, hence the lower the total pressure against, and the wear, on, the sides of the grooves.

The radial thickness determines the spring tension against the cylinder walls. So long as this is above a certain figure there is no object in increasing the thickness, which merely involves an increase in friction and wear on the cylinder walls.

In a well-made ring a spring pressure of from 5 to 6 lbs. per square inch is sufficient, and any further pressure results merely in extra friction without any compensating advantage. The spring tension required, however, depends to some extent upon the amount of clearance between the "lands" of the piston and the cylinder walls. If clearance is too great, a considerable area of the side of the ring may be explosed to the full gas pressure, and the ring may be pressed so hard against the lower face of the groove that its spring tension will not overcome the friction against the side of the groove.

The top land should always be as deep as possible, to provide adequate protection for the top piston ring.

Difference of opinion both as to the most desirable stiffness of spring tension and as to the width of the ring. Stiffness depends much on accuracy of workmanship. When workmanship is of high order; a spring tension of 4 to 5 lb. per square inch is sufficient.

As to the width of the ring, there can be little doubt that the narrower it is the better, the area of the surface in the ring-grooves is governed solely by the radial thickness, while both the inertia and the friction drag of the rings is directly proportional to the width. The loadings on the sides of the groove and the wear will increase directly in proportion to the width of the ring. Wear in the ring grooves is one of the most troublesome features; once a ring becomes slack in its groove its reciprocating motion therein converts it into an efficient pump for pumping oil into the combustion chamber.

In practice the minimum width of a piston ring is determined largely by manufacturing considerations and its fragility in handling and generaly speaking it should be as narrow as is consistent with ease of handling and of manufacture.

The Connecting Rod

Functions of the connecting-rod.

Ratio of connecting-rod length to crank length. Large ratio reduces friction, wear, and vibration; short rod gives a lighter motor, over-all dimension of engine is lower.

Important that piston-end shall be as light as possible. The connecting-rod is almost invariably machined; little doubt that I section is the best, but expensive. Bolts of the big-end are a source of weakness; a certain amount of deformation of the big-end bearing generally takes place, and severe stresses are thrown upon the bolts; moreover, continual reversal of stress tends to cause the metal to become brittle; generally advisable to renew, or to anneal the big-end bolts from time to time. In high-class engines it is customary to forge the big-end bearing cap in one piece with the rod, and afterwards part it through the center. There have been many accidents due to broken big-end bolts with marine type of bearing.

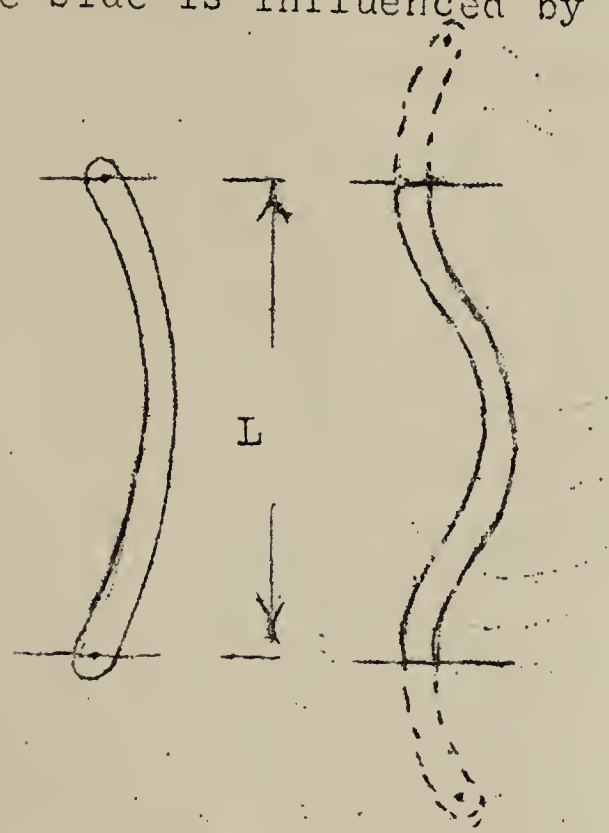
Chief considerations in design of connecting rod are:

(1) That it shall be stiff enough to resist not only bending and crushing, but also vibration.
(2) That it shall be as light as possible.
(3) That the big-end of the rod shall be sufficiently rigid to ensure adequate support to the bearing.

To fulfill conditions (1) and (3) means that the section of the rod must have an ample margine of safety. When I section rods are used, the section must have sufficient width to resist vibration in the plane of the crankshaft, as well as sufficient depth to resist bending, - many troubles due to vibration sideways.

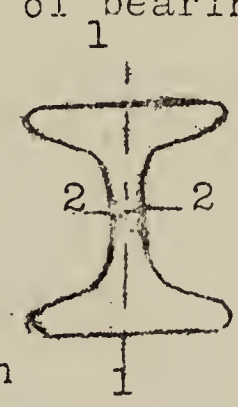
The connecting-rod is subjected to alternating tensile and compressive stresses, and also to flexural stresses due to centrifugal forces. Greatest tensile stress occurs at head-end of stroke, due to inertia force. At crank-end dead-center the stress is compressive. The explosion pressure of the gas within the cylinder puts a compressive load on the connecting-rod which, at ordinary speeds, is greater than the compressive force due to inertia at the lower dead-center.

In the design of the connecting-rod body we should first calculate the cross-section with respect to the compressive forces, and then investigate for the unit stresses under the tensile and the flectural loads. The direct axial compressive stress is increased by the tendency to lateral deflection which produces a flexural stress, adding to the axial stress on one side and deducting from it on the other side. This inclination to spring to the side is influenced by the condition of the ends.



In the plane of rotation, the connecting-rod is free to swing about the journals and may assume the shape in diagram 2. When the ends are restrained, the column takes the form of a compound curve shown in diagram 1. This is the case of the connecting-rod in the plane of the bearing axis.

Fixing the ends quadruples the resistance to buckling. Hence, moment of inertia of the section in the plane of rotation would be four times as great as moment of inertia in plane of bearing axis.



To produce the same fiber stress in the rod about the two normal axes of the section, it is necessary to make the radius of gyration in the plane of rotation twice as great as the radius of gyration in the plane of the bearing axes. The width of the flanges must be determined to make the moment of inertia about axis 2-2 four times as great as about axis 1-1.

Little advantage results from use of a tapered rod; for stress by column formula is not much greater at the center than at the ends in direct compression.

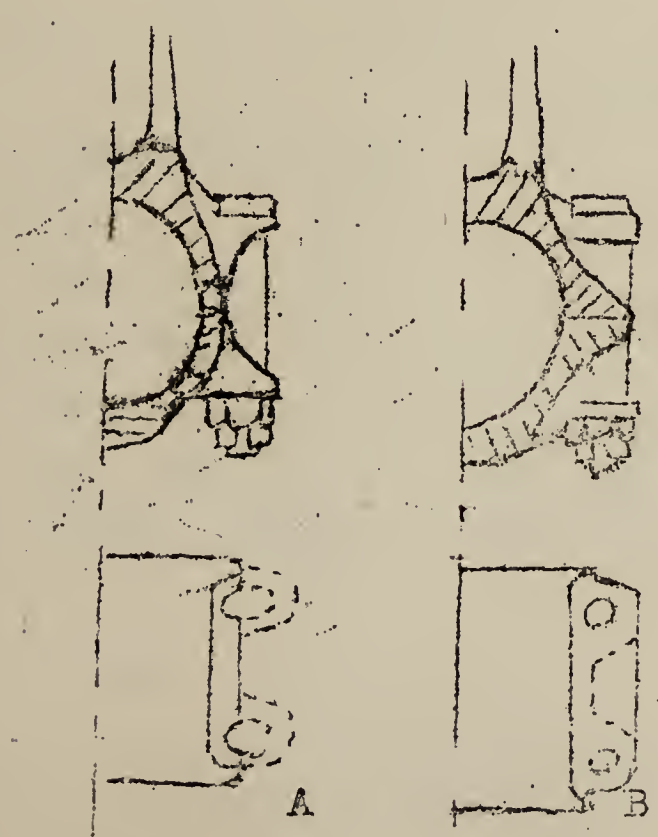
Tubular section has advantage of being easily machined and of permitting use of a higher working unit stress.

The most economical form, in the matter of material, is the I section.

It is most important to keep the weight of the connecting-rod as low as possible consistent with fulfilling other requirements. In many cases it is quite as important to keep down the weight of the rotating end as the reciprocating end. In a six-cylinder engine the rotating weight plays a more important part than the reciprocating weight; in a four-cylinder engine it is the reciprocating weight of the connecting-rod which must be first considered.

In the design of the big-end the first consideration is to obtain a uniform support for the bearing; and not only must the eye of the rod be made as rigid as possible, but also the load transmitted down the rod must be distributed over it as uniformly as possible. Useless to provide a wide bearing because of practical impossibility either of obtaining the necessary rigidity or of distributing the load over it. In Ricardo's opinion, it is doubtful whether any useful bearing surface can be obtained when the width of the big-end bearing exceeds the diameter of the crank-pin; in any case it is desirable to radius-out the outer webs of the shank at their junction with the big-end in order to distribute the load better.

Another source of weakness is lack of sufficient abutment for the two halves of the connecting-rod bearing,- the eye tends to close-in and nip the crankpin at the sides. Figure A shows a design of big-end which gave rise to constant failures of crank-pin bearings. Figure B shows the same rod after the design had been corrected, - no further trouble arose.



Problem of fitting the bearing in the connecting-rod big-end has always been a difficult one. Most reliable method is to run-in the white metal. There are practical objections to this method:

(1) In the event of the lining being run-out, it is not easily replaced.
(2) If the lining runs-out, the steel surface of the rod will come in contact with the crankpin and may do considerable damage.
(3) When high tensile steel rods are used, it is difficult to get the bearing metal to adhere uniformly, and it is not easy to ensure good metallic contact.

However, direct white metalling of the rod itself, is probably the only satisfactory method for high-duty engines. Alternatives are die cast white-metal bearings or white-metal-lined bronze shells. The former are suitable only for light duty; also they are open to the objection that if the white metal runs-out the steel of the rod will come in contact with the crankpin; for medium-duty, bronze shells lined with a thin lining of white metal offers the most satisfactory compromise. The white-metal lining should always be thin, from 1/32" to 1/16" is ample.

Materials. The metal in most common use is carbon steel, heat treated. For high-grade motors a steel of chrome-nickel or chrome-vanadaium alloy, also heat treated, is sometimes employed.

To fulfill the requirements as regards rigidity, necessitates an ample margin of safety; there is seldom occasion to call for high tensile alloy steels. Recently the use of special aluminum alloys has been tried, and has been found satisfactory in several engines. The advantages of aluminum connecting-rods are important, not only is the weight of the rod reduced to less than half that of a steel rod, but its thermal conductivity is four or five times greater. Before we can be satisfied definitely that the aluminum rods are uniformly reliable, it will be necessary to have experience of the prolonged use of many hundred of rods.

Typical Connecting-Rods.-

Curtiss OX. Cylinders in the two rows are staggered so that the big ends of each pair lie side by side on the same crankpin. The rods are much the same as for a vertical engine. (Illustrate).

The more common practice with V engines is to place the cylinder of each pair opposite one another which puts the connecting-rods into the same plane. With the Liberty, Hispano-Suiza, and others, each

pair of rods consists of a plain rod and a forked rod. The forked rod clamps the big end bronze bushing; the plain rod works on the outside of this bronze bushing between the forks of the forked rod. Difficult problem to make the twonrods of different form exactly alike as regards weight, center of gravity, and radius of gyration. (Illustrate each).

A master rod is used and a shorter rod is attached to a pin which is held in the big end of the master rod.

In W engines the articulated rod is most common. The Napier "Lion" uses a central master rod, on each side of which is mounted an articulated rod carried on pins fixed in lugs integral with the big end of the master rod. Character of motion.

The articulated arrangement suffers some disadvantages. Explosion pressures transmitted along the short rod impose stresses on the master rod which under unfavorable conditions may be serious.

Conclusion. Connecting rods must be as light as possible consistent with rigidity and strength, in order to reduce the inertia forces of reciprocation and rotation. The crankpin bosses require adequate reinforcement to prevent distortion with consequent binding of the bearings under heavy pressure. The liners must have intimate contact with the rod to facilitate the flow of heat from the bearing surfaces. The bores of the wrist-pin and the crankpin bosses must be parallel with each other.

Light weight and uniform section is secured by machining the connecting rod all over. Both the tubular and the I sections are popular. The ratio of connecting-rod length to crank varies between 3.4 and 4.5 with an average of about 3.6.

The Crankshaft

The crankshaft is subjected to three forces - the gas pressure on the piston, the inertia force of the reciprocating masses, and the inertia force of the rotating masses. These forces set up variou stresses- compressive, tensile, flexural, and torsional in different directions and in variable proportions during the cycle.

The proportioning of the crankshaft is characterized by greater diagreement among designers than that of any other part of the engines. Different makes use crankshafts of which the diameters vary by as much as 40 per cent for engines of the same leading dimensions.

The actual dimensions are generally fixed, not by the strength required, but by such considerations as bearing surface and rigidity. Tendency toward higher speeds and higher mean gas pressures, has a decided influence upon crankshaft development. Journal diameters have increased steadily with the need for greater rigidity while journal lengths have shortened with improvement in bearing lubrication.

Generally speaking, if the crankshaft is large enough in diameter to be free from whip, and to provide the bearing surface necessary, it will have more than sufficient strength. To provide adequate bearing surface the projected area of the crankpin should, be at least 30 per cent of the area of the piston for a piston speed of 1000 ft. per minute, and the diameter of the pin will be equal to nearly half that of the piston, which is considerably greater than is needed to resist any bending stress. Much depends upon both the speed and the method of lubrication. It is advisable, in order to avoid torsional vibrations, to employ still heavier crankshafts.

When the rubbing velocity exceeds about 8 ft. per second - the load factor only, that is, the product of load and speed, need be taken into account; the maximum pressure, so long as it is not high enough to distort or crush the bearing material, is of little moment.

<u>Limiting Load Factor</u>.- The highest load factor which can safely be carried by a bearing depends upon:

(1) The degree of lubrication.
(2) Both the viscosity and the oiliness of the oil.
(3) The facilities for conducting away the heat generated in the bearing.

Under the most favorable circumstances, load factors as high as 20,000 lb. ft. per second can be safely carried in a journal bearing with alternating load.

Where the rubbing velocity is low the only limit to the maximum load is set by the rigidity of the members. There is no danger of the oil film being broken down by pressure alone, provided

there is scope for replenishment. When serious distortion takes place, the load factor will be increased locally causing rapid local heat flow and ultimate breakdown of the oil film.

From point of view of reducing the load factor the requisite area of bearing surface should be obtained by lengthening the journal rather than by increasing its diameter; since any increase in diameter involves a corresponding increase in rubbing speed. On the other hand, from the point of view of rigidity and freedom from distortion a long journal is very undesirable, since the load cannot be evenly distributed over it, nor can so much cooling oil be circulated through it. With loads that obtain in the bearings of a high-speed engine, the pressures are so high that it is seldom practicable on the score of distortion to make a journal wider than one and a half times its diameter. Generally speaking it is desirable so to arrange the main bearings that the width of the journal is about equal to the diameter.

Material. The metal from which the crankshaft is constructed should be tough enough to withstand shocks and hard enough to resist wear.

Since the crankshaft should not deflect greatly underload, either laterally or in torsion, the diameter is usually not based upon the permissible working stress, but upon the modulus of elasticity. On the other hand, a high factor of safety is recommended in order to diminish the probability of fracture by fatigue. The working fiber stress is therefore kept low to reduce the distortion while the ultimate strength is given a high value to prevent fatigue.

There is always a considerable difference of opinion as to the most suitable material for crankshafts. The two essential requirements are resistance to fatigue and surface hardness. Since it is rigidity which largely controls the design of a crankshaft, all steels are nearly on a par in this respect. In Ricardo's opinion a straight carbon steel fulfills all the requisite conditions quite satisfactorily for most purposes except for aircraft engines. These engines are excepted because:

(1) Having no flywheel, aircraft engines are less susceptible to torsional oscillations in the crankshaft, and the shafts can be made considerably lighter.
(2) In order to save weight in the shaft and in the design generally, a bearing is always fitted between each crankthrow, hence the tendency to whip is greatly reduced.
(3) Owing to their relatively short working life, the area of bearing surface is generally cut-down in order to reduce weight.

For these reasons, the crankshaft for an aero engine may be made relatively much lighter than for other types, and, as a result, it is much more highly stressed.

Fracture of a crankshaft of reasonable design is a rare occurrence, and is generally due either to want of adequate fillets at the crankpins and journals, or to periodic torsional vibration; either of these causes will result in eventual fracture from fatigue no matter what the material may be.

It is particularly important that the fillets at the junctions of the crankpins and journals with the crank webs should be of adequate size, - not less than 3/8 in. radius.

The important consideration today is to reduce the rate of wear.

Wear is generally due to abrasive action of particles of grit carried by the oil; therefore, the rate of wear depends upon the thickness of the oil film, which, in turn, depends upon the pressure and temperature.

Rate of wear also depends upon the hardness of the surfaces of the materials. It is desirable that the difference in hardness between the two surfaces should be as large as possible.

The once popular idea that a hard white metal should be used is erroneous, - the surface of the white-metal should be as soft as is consistent with the necessary resistance to crushing. The worst results are found when two similar and relatively soft materials are used for the two members of a bearing.

In the case of hardened steel, both surfaces are so hard as to be very little affected by grit; but, in the event of breakdown of the oil film, the two are liable to become welded.

While the greatest wear generally occurs in the main journal bearings, actual failure of the bearing material is more frequent on the crankpins, even when both journal and crankpin bearings are subjected to the same load factor. The greater liability to failure of the crankpin bearings is due to:

(1) The smaller heat capacity which the bearing has to draw upon.

(2) Inadequate support of the bearing in the connecting rod.

In the event of a temporary stoppage or slowing-down of the oil circulation, the crankpin bearing will attain a much higher temperature than the main journal bearings because the latter have behind them the heat capacity of the whole crankcase.

With regard to wear, in most lubrication systems the oil is delivered first to the main journal bearings, and passes thence to the crankpin bearings, with the result that the journal bearings receive and retain most of the grit.

Rate of wear appears to be almost directly proportional to the load factor on the bearing, and the surface hardness of the shaft, and it is almost always most rapid at the bearing which receives its lubrication first.

The advantage to be gained by the substitution of ball bearings is a trifling one from the point of view of mechanical efficiency; their real advantage lies in the fact they require very little lubrication.

The great objection to the use of ball or roller bearings for crankshaft and connecting-rod bearings is the difficulty of getting them into position.

It has been pointed-out that the bearing loads at high speed are due mainly to inertia forces, which may exceed the explosion pressure. The resulting mean bearing pressure is very high and, unless some means are used to cool the bearings and the oil, the oil will not have a high enough viscosity to separate the bearing surfaces. To overcome this difficulty, a method analogous to the water-cooling system is often used. The oil reservoir is sometimes cast with cooling fins to dissipate the heat; an oil

Examination

Airplane Engine Design

December 14, 1923.

1. What would be the thermal efficiency of an engine that gives a maximum temperature of 2800° F., temp. at exhaust 1200°F, at end of compression 600° F., if temperature of atmosphere is 50°F.?

2. (a) What are the advantages of high compression?
(b) What are the objections to high compression?

3. If ignition is 30° in advance of dead-center when engine is running 1500 r.p.m. (a) at about what crank angle would the maximum explosion pressure be reached? (b) at about what crank angle would the combustion be complete?

4. What would be the percentages of CO_2, CO, and O in exhaust gases if ratio of air to gasoline in the mixture is 17 to 1?

5. (a) The exact timing of which of the four valve-events is most important in a multiple-cylinder engine supplied by one carburetor?
(b) If valves are set for giving maximum power at high speeds, why will the engine not run as well nor be as flexible at low speeds?

6. What would be the temperature at the end of the suction stroke if compression ratio is 5 and the temperature of the exhaust gases is 1200° F., the air begins at 50° F?

7. (a) About what ratio of air to gasoline in mixture gives the highest thermal efficiency?
(b) What mixture strength gives complete combustion?

8. If the temperature of the inlet charge is 155° F., what would be the ideal volumetric efficiency?

9. What would be the velocity of inlet flow if the suction pressure is - 1.5 lbs.?

10. If an engine is running at 1200 r.p.m., about what would be the difference in the weight of charge (lbs. per 1000 rev.) for engine hot and engine cold?

11. If it is found that the exhaust temperature is 765° C for a mixture strength of 14.5, about what would be the exhaust temperature for a mixture strength of 12.; for 18.?

12. If it is found that the mean negative pressure in the inlet manifold of a four cylinder engine is 1. lb. at 800 r.p.m., about what would you expect it to be at 1800 r.p.m.?

13. If an engine develops 162 H.P. at 1500 r.p.m. when exhausting straight into the air, (a) about what H.P. would be developed if suitable exhaust pipes are used? (b) about what H.P. if an efficient silencer is added?

14. (a) About what percentage of the total mechanical losses is the pumping loss?
(b) What is the most important of all frictional losses?
(c) About what percent of the total mechanical losses is the power to drive all the auxiliaries, - (circulation, pump, oil parts, magneto, etc.)?

15. About what drop in pressure during the suction stroke would there be (a) for a gas velocity of 125 ft. per sec.? (b) for 200 ft. per sec.?

16. If a Liberty-400 gives 390 H.P. at 1600 r.p.m., what is the torque at that speed?

17. The weight of a 400 H.P. air-cooled static engine is 600 lbs. The weight of a 400 H.P. water-cooled engine (dry) is 800 lbs. What would be the weight of each power plant with fuel, etc. for a 1 hour flight, and for a 10 hour flight?

18. (a) What are the advantages of air-cooled engine over water-cooled?
(b) What are the disadvantages?

19. (a) How are the piston-pins usually lubricated?
(b) How are the cylinder walls lubricated?

20. (a) What should be the location of the oil relief valve?
(b) What should be the location of oil filters?

21. What is the chief requisite of cam shafts?

22. What is a desirable proportion of nose of tangential cam with reference to diameter of base circle and of roller?

23. What spring force would be required for a tangential cam which gives a lift of 3/8" to a 2" valve when engine makes 1600 rpm? Radius of base circle is 3/4", radius of nose is 3/8". The roller is 1" diam. and the weight of valve and parts (including one-half of spring) is 1-1/4 lbs.

24. What diameter of wire would be suitable for the spring in question 23 if the spring is 1-1/2" diam.

25. What would be the number of free vibrations of a valve spring of 50 lbs. rating weighing 3/8 lbs.?

26. What are the objections to use of relatively large valves?

27. For what reason is a domed cylinder head with inclined valves better than a flat head with vertical valves,

28. For what reason do some designers cut-away the sides of the piston skirt?

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$dR_t = dM\,dr$ — $\frac{y_0}{r}$ $\frac{x_t}{r}$ $x_t \frac{y}{r}$ $\frac{y}{r}$ $dM\,dx$ $\int \frac{x}{r} dM = 0$

$y\,dM$ — $x_0 M = M x_0$

$\frac{y}{r}$

(4) $dR_n = \frac{x}{r}\,dR_x = \int \frac{x}{r}\,\omega^2 r\,dM = \omega^2 x_0 M$

$y\,\omega^2 M = \omega^2 y_0 M = 0$

$\frac{y}{r^2}$

$r^2\,dM = I_n$